INDICATOR DIAGRAMS.

A TREATISE

ON THE USE OF THE INDICATOR

AND ITS

APPLICATION TO THE STEAM ENGINE.

BY

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PRICE 6/- NET.

THE SCIENTIFIC PUBLISHING CO.,
MANCHESTER.
1899.

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PUBLISHING CO.
MANCHESTER.

PREFACE.

THE usefulness of the Indicator as a steam engine analyser is now universally acknowledged, and the engineer who has to either make or use an engine must feel at a disadvantage if he has not some knowledge of how to analyse its working by means of this instrument.

The complete story, as written by the Indicator pencil, ofttimes can only be interpreted by the assistance of much experience, together with clear ideas as to the capacity of the instrument to record the truth. In the following pages the author has endeavoured to give the result of a certain amount of experience as well as to indicate in what direction to look for those irregularities commonly found under ordinary conditions of use.

The author has been much assisted by Mr. J. G. Hudson M.I.C.E., of Messrs. Hick, Hargreaves, and Co.; by Mr. Edward G. Hiller, M.I.M.E., chief engineer of the National Boiler and General Insurance Company, of Manchester; and by Mr. W. H. Fowler, Wh. Sc., A.M.I.C.E., M.I.M.E., who kindly placed some useful examples of Indicator Diagrams at his disposal. He also wishes to acknowledge his indebtedness to the several manufacturers who have favoured him with illustrations; to the Proceedings of the American Society of Mechanical Engineers, especially for some valuable matter in connection with the testing of Indicators by Professors Carpenter and Jacobus; and to the work of Professor Osborne Reynolds, F.R.S., and Mr. Brightmore, B. Sc., in the Proceedings of the Institution of Civil Engineers.

W.W.F.P.

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CHAPTER I.

MEASUREMENT OF POWER WITH THE INDICATOR.

The Indicated Horse Power, or the rate at which an engine receives mechanical energy from the driving agent (steam, gas, &c.), is measured by the mechanical energy imparted per minute in foot-pounds, divided by 33,000. It is only convenient to apply an indicator to engines in which the cylinder is rigidly attached to the framing or bed plate, and on this account such motors as steam turbines and similar engines cannot have their mechanical efficiency measured.

The work done per stroke of L feet by the steam or other pressure agent on a piston whose area is A square inches is

$$p\,A\,L \text{ foot-pounds,}$$

p being the mean effective pressure per square inch in pounds.

If N is the number of *useful* strokes* per minute, *i.e.*,

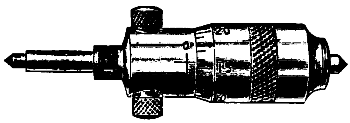

FIG. 1.

the number of strokes of the piston when the pressure p is acting; then the *indicated horse power*—

$$\text{I.H.P.} = \frac{p\,L\,A\,N}{33000}$$

This applies to any and every cylinder when the engine is working approximately uniformly.

When the driving agent can find its way to both sides of the piston, it is necessary to find the indicated horse power for each side separately, and add the results together for the total horse power.

The quantity N in the above expression may be obtained by some form of tachometer, or by counting and a stop

* In a gas or oil engine N will represent the number of explosions per minute.

watch, or by some form of counter as shown in a previous chapter. The area of the piston is best obtained by removing the cylinder cover and measuring the diameter of the cylinder with a micrometer gauge. One of the gauges, for small diameters, is shown in Fig. 1, made by the L. S. Starrett Company, of Athol, Mass., U.S.A. It reads to thousandths of an inch if necessary, and is used like a micrometer caliper. Another gauge by the same makers, for large work, is shown in Fig. 2, and a cylinder of any diameter could be gauged by one of these, by simply putting on enough lengthening pieces.

Fig. 2.

It is necessary to measure the diameter of the cylinder and piston rod at a number of positions, and take the mean, as these often wear out of truth.

The length of the stroke can be obtained by marking the cross-head guide at the extreme ends of the stroke.

The mean effective pressure is obtained from the indicator diagram by means of an averager as previously described. The experimenters will find it convenient to record their observations on a blank log form, such as the one given below, a separate form being required for each cylinder.

If in the above form the revolutions per minute are not read direct, but are obtained from periodical readings of a counter, then the counter reading should be inserted in another column and the corresponding number of revolutions inserted in the speed column. The different indicators in common use, with one exception, work upon the same principle, though their details exhibit considerable variation.

LOG OF TEST FOR INDICATED HORSE POWER.

Date.................................. Observers..

Kind of engine.. Maker's number.........

Maker's name... Work done by engine...

Cylinder to which the following data refer...

Diameter of cylinder...........inches. Diameter of piston rod...........inches. Stroke...........inches.

Name of indicators.. Number of spring..

Cut off—Crank end.. Head end...

Speed determined by... Barometer...................................

$I.H.P. = CPN$. $C =$ ....Crank end. $C =$ ....Head end.

NOTE.—**In the case of a gas or oil engine, the number of explosions per minute must be substituted for revolutions in the third column.**

Time.	Boiler or separator pressure, lbs. per sq. in.	Revolutions per minute.	Crank end.			Head end.			Total I.H.P.
			Number of diagram.	Mean effective pressure, lbs. per sq. in.	I.H.P.	Number of diagram.	Mean effective pressure, lbs. per sq. in.	I.H.P.	

The Thompson Indicator.—This instrument is made by Messrs. Schäffer and Budenberg in two sizes, the larger of which is shown in section in Fig. 3. The indicator is attached to the cylinder by means of the indicator cock A, which permits of the indicator being put on or removed

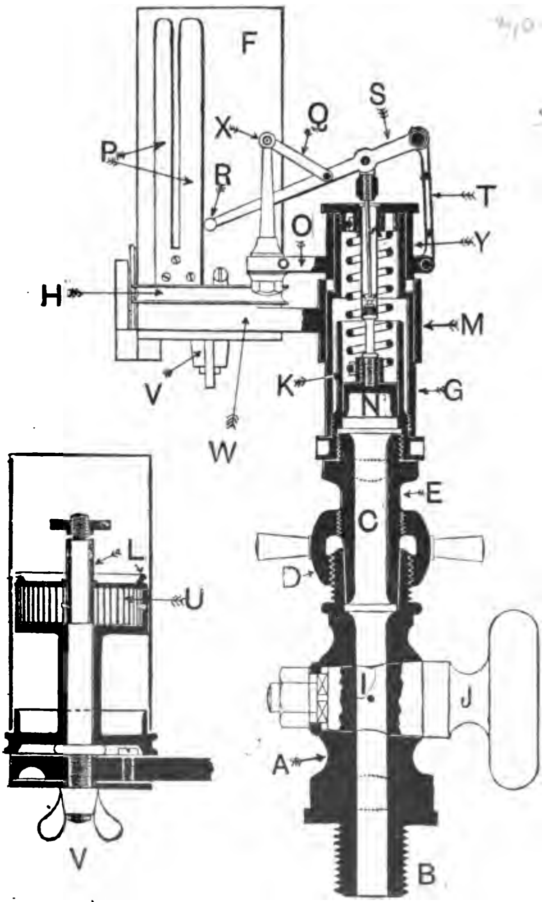

FIG. 3.—The Thompson Indicator.

without stopping the engine. The indicator cocks, as made in this country, are all screwed at B, similar to the Whitworth standard ¾-in. bolt, 10 threads to the inch, and indicator bosses are correspondingly tapped, so that any

instrument may be used on any engine. The indicator cylinder K is screwed on to the distance piece E, which is secured to the indicator cock A by the hand nut D. The lower end of the distance piece E is made to fit into a conical socket, and the metal to metal joint is rendered quite gas-tight by the great axial pressure produced by the compound screw in D. The channel C is very carefully bored to half the sectional area of the cylinder K, so that with a supplementary piston having a longer piston rod, the same spring may be used for two different pressures, or in internal combustion engines the smaller piston may be used with a stiff spring so as to avoid as far as practicable the effects of inertia.

The piston N is ground to slide in the cylinder K without leakage and without any appreciable resistance. It is made of gun-metal, and very light. The piston rod is screwed into a boss on the top side of the piston, and is made hollow for about two thirds of its length, its upper end sliding freely in a boss on the under side of the cylinder cover, which acts as a guide.

The spiral spring is securely attached to a solid head-piece at each end, which are screwed respectively on to the piston boss, and the cylinder cover boss.

The pencil R is situated in the end of the pencil lever S, which can rotate about a pin through its right-hand end, the pin being secured in the upper end of the pendulum link T, which can vibrate about a pin fixed in lugs on the loose collar Y. Another lug O, on the same collar, carries the small pillar whose upper end holds the pin X, about which the controlling link Q rotates.

It will be noticed that if the right-hand end of S move in a horizontal straight line, and the length of Q were half the length of S, then we should have Scott-Russell's straight-line motion in which R would move in a vertical line behind the centre line of the pillar X. But as the lever S is longer for the purpose of multiplying the movement of the piston, then Q has to be made correspondingly shorter to keep R's motion in the same vertical line. The motion of the upper end of the link T is approximately in a straight line. The lever S is connected to the piston rod by the small connecting rod shown in the figure, the lower joint being a double ball joint. This gives perfect freedom of motion, and allows of any wear being taken up.

The outer casing G of the indicator cylinder has rigidly

fixed to it a collar M, which carries the arm W, which supports the paper drum F. A section of this drum is given on the left of the figure. It is very light, to avoid inertia effects, and is rotated in one direction by a cord round the grooved pulley H, motion in the opposite direction being obtained by the coiling up of the clock spring U. The thin paper drum F slides tightly on the projecting rim of the cord pulley, its exact position being determined by a slot passing over a little screw shown in the upper figure. The coiled spring U is attached to the hollow spindle of the cord pulley and to the long collar L which can turn round the fixed central spindle when not held secure by the small thumb nut at its top end. The butterfly nut V, at its lower end, is for the purpose of securing the cord guide pulley and arm, but this is better seen in succeeding illustrations.

In this indicator the motion of the pencil is four times that of the piston, the maximum height of diagram being 3 in. The pencil is brought into contact with the paper on the drum by bringing round the arm O; this can also be better seen in later figures. The paper is held on the drum by the clips P, and great care must be exercised to prevent any moisture from dropping on the drum, as it will act upon the starch of the highly-glazed surface of the paper and stick it to the drum, causing much inconvenience and perhaps delay. The little screws which secure the clips should not be touched unless absolutely necessary, as they are so short as to be easily dropped and lost; also frequent removal will make them loose in their sockets, and then the clips will not hold the paper tight on the drum. The paper must be carefully placed on the drum. This may be accomplished by making a right angle bend in the paper by creasing one end about a quarter of an inch from the edge. Hitch the bend in the longest clip, and bend the paper round the drum, hitching the other end under the other clip. Hold both ends tightly with finger and thumb of one hand, and with the other ease the paper on to the drum. In this way the paper will slide smoothly over the drum into position. The paper must not bulge at all, but fit tightly all over the drum, otherwise the pencil gear may be broken by tearing the paper in the region of a bulge.

The indicator piston and cylinder should be oiled with a pure mineral oil both before and after use, especially when used on a gas engine.

The pressure of the pencil on the paper is regulated by screwing the little screw stop in or out. A heavy dark outline to the diagram is not desirable, but a clear line only dark enough to be plainly visible. Too dark a line means that the pencil presses too heavily on the paper, causing too much friction, and consequent distortion of the diagram.

With the large Thompson indicator the makers recommend a spring so selected that

number of spring = ·37 maximum pressure + 6,

and with the small high-speed instrument

number of spring = ·55 maximum pressure + 8.

The springs usually made for the large indicator are numbered 8, 10, 12, 16, 20, 24, 30, 32, 40, 48, 60, 72, and 80, while those made for the small instrument are 12, 16, 20, 24, 32, 48, 56, 64, 80, 90, 120, 160, 200, 240. The numbers are expressed by some makers in the form of fractions, thus $\frac{1}{8}$, $\frac{1}{10}$, $\frac{1}{12}$, &c., instead of 8, 10, 12, &c., the meaning being, of course, that the 1 in. of movement of the pencil represents 8 lb., 10 lb., or 12 lb. per sq. in.

A Reducing Gear.—The motion of the circumference of the pulley cord H is not more than 7 in., and consequently the cord can seldom be attached direct to the crosshead, but will require some mechanism to reduce the motion of the piston and crosshead down to that of the indicator cord. One of these reducing gears is illustrated in Figs. 4 and 5 a very handy mechanism by the same makers as the above indicator. The indicator is the small high-speed type and carries below the cord guide pulley arm, a second arm shown shaded, clamped by the same screw. This arm carries the large and small cord pulleys. The large pulley receives the cord direct from the crosshead, and the little pulley cord is connected direct to the indicator drum. The ratio of the diameters gives the amount of reduction. The small pulleys are made in considerable variety for different reductions, and can readily be changed to suit the stroke of an engine. The large and small pulleys can be quickly thrown in and out of gear, for starting or stopping the paper drum. This will be found to be exceedingly convenient in practice. The large pulley has also a coiled spring on its boss to rotate it during the inward stroke. The whole arrangement is neat and compact, and does not require any special fixing on the engine. A plan of the

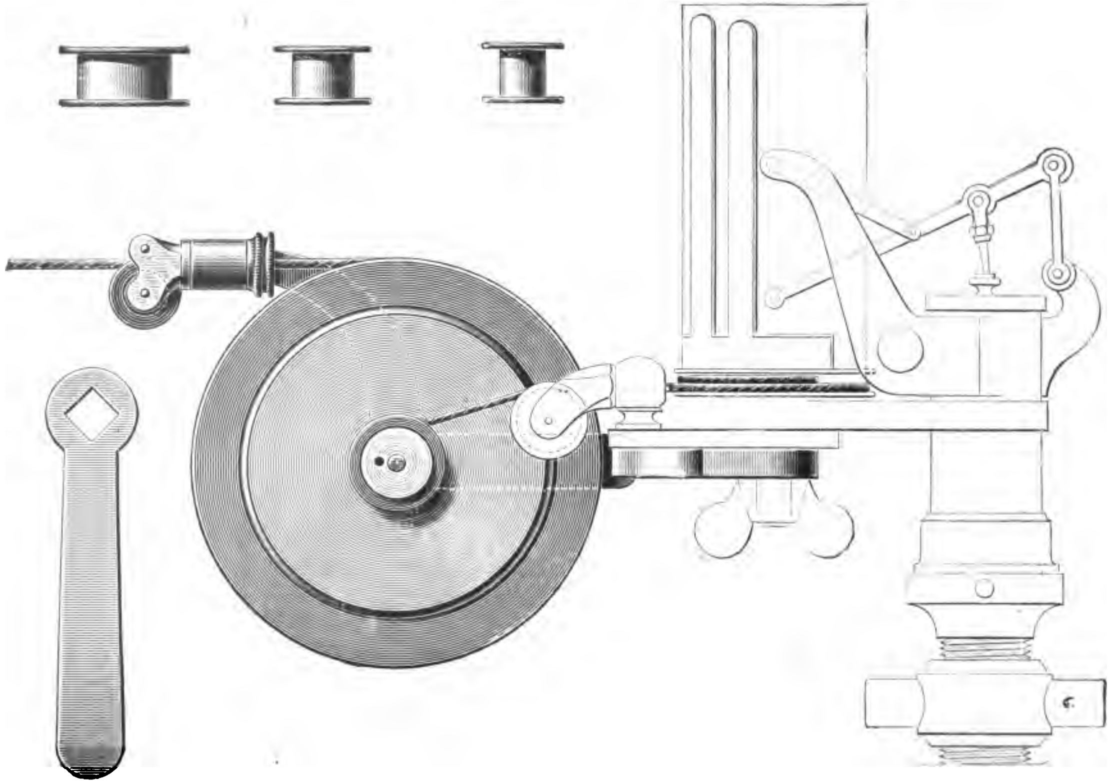

FIG. 4.—SCHAFFER AND BUDENBERG'S SELF-CONTAINED REDUCING GEAR.

reducing gear is shown in Fig. 5. The pencil mechanism of the indicator is the most delicate part of the instrument, it being made very light to avoid the inertia effects. It should not be kept working longer than is really necessary, so as to prevent the joints from wearing, thus prolonging the life of the instrument. When cleaning or changing a spring, all screwed parts should be well screwed home, though not too tightly, otherwise it may be difficult to unscrew them when it may be necessary to do so during the time when the indicator is hot.

Abrasive material such as flour of emery should never be allowed to come near the indicator, as a little grit may render the instrument worthless for careful work.

The plug J of the indicator cock A should be carefully ground in whenever it shows signs of leaking, and this will be not uncommon with internal-combustion engines. In the plug J will be noticed a small hole I, there being a

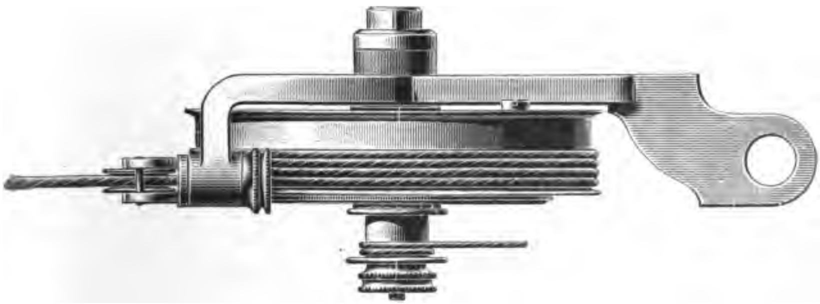

FIG. 5.—Plan of Reducing Gear.

corresponding hole in one side of the cock. This is absolutely necessary when the cock is attached to a steam engine to get rid of the condensed steam, otherwise the indicator will register the effect of the inertia of a certain amount of water, as well as the steam pressure, and thus a false diagram will be the result. The drain hole should be opened previous to taking a diagram, and until blue steam comes from it, at which time the indicator and connections will be uniformly hot. When the indicator cock is put on an engine, near which a number of students or other people may have to work or pass, care should be exercised to arrange the position of this drain hole, so that the condensed steam from it will not scald anyone; or if the indicator is placed near or over a brake wheel, the

condensed steam must not be delivered on to the wheel, as it would instantly alter the friction of the brake, and perhaps cause much unpleasant oscillation of the brake strap.

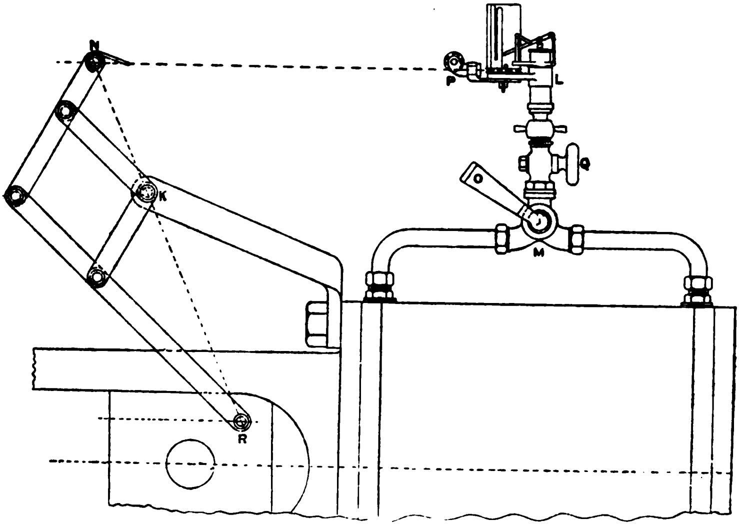

Fig. 6.—Method of Rigging up a Single Indicator on a Double-acting Cylinder.

Cord of too large a diameter should not be used, as there may not be room for two or more pieces side by side on the cord drum, and then one will ride over the other and the motion of the drum will not be proportional to that of the piston.

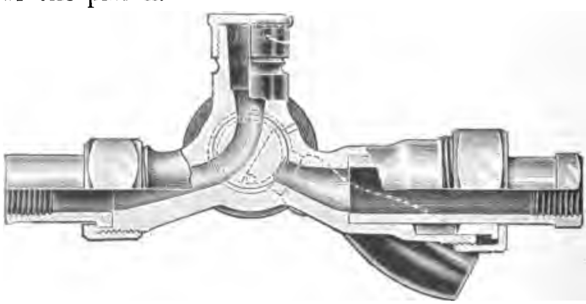

Fig. 7.—Three-way Cock for Indicator Rigging.

Special cord should always be used, as it stretches very slightly. It is very expensive, but is almost indispensable for correct diagrams, especially at high speed.

CHAPTER II.

INDICATOR RIGS AND REDUCING GEARS.

Indicator Rigs.—With slow running engines a single indicator may be used, connected to both ends of the cylinder through a three-way cock, as shown at M in Fig. 6.

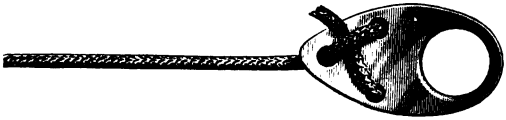

FIG. 8.—Cord Adjuster.

The handle O to the three-way cock has a couple of lines cut upon it, indicating the direction which the steam is taking through it, and by its use is determined to which end of the cylinder any particular diagram belongs. A

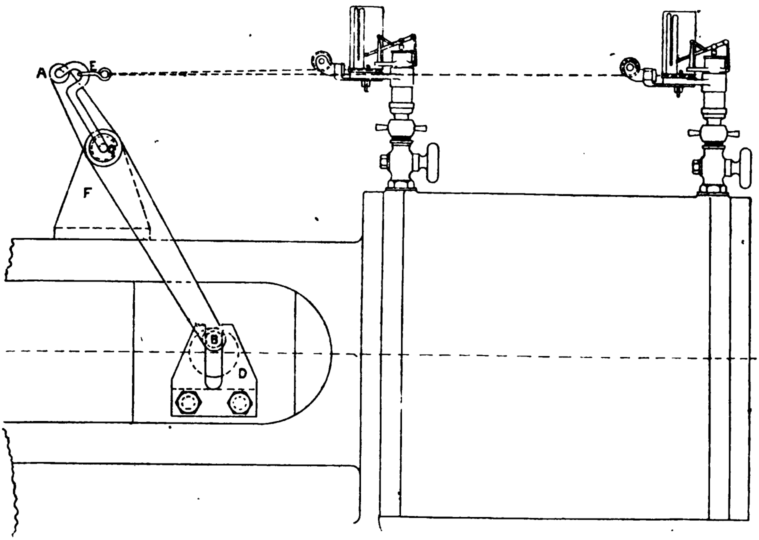

FIG. 9.—Method of Rigging up a Pair of Indicators on a Double-acting Cylinder.

section of this cock, as supplied with the Tabor indicator, is shown in Fig. 7. The pipes connecting this cock to the cylinder should be of copper, and with easy bends. The channels along which steam has to pass must be free from obstructions, and large enough to prevent wire-

drawing. It is generally more satisfactory to use the ordinary indicator cock Q, Fig. 6, as well as the three-way cock, as some of these are not provided with water escape holes. The indicator cord adjuster, Fig. 8, will be useful to anyone who may have to carry the indicator about to different engines, or use it on different sizes of the same class of engine.

With high-speed engines two indicators are necessary, as shown in Fig. 9, to obtain faithful diagrams.

A single instrument has been used by some experimenters, but it is now recognised that in general two should be used. To prevent the indicator cord of the head end indicator fouling the other instrument, it should be set at an angle with the axis of the cylinder, as shown in Fig. 10, both cords being attached to the same hook

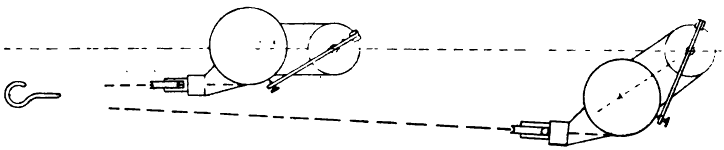

FIG. 10.

Sometimes the indicators are fixed to a short bend screwed into the cylinder covers. This secures that the piston does not cover up the channel to the indicator at the end of the stroke. If the engine is of the vertical type, a short bend may connect up the indicator and three-way cocks, so as to get the indicator vertical, but this is not a ecessity.

Reducing Gears may be divided into two classes—(1), those which are geometrically perfect, that is, give an exact reproduction of the motion of the piston to a smaller scale, and (2) those which are approximately correct.

The pentagraph arrangement shown in Fig. 6 is one of the former group. It consists of a series of parallel rods pinned together, with one intermediate point K fixed; then the motion of N is a duplicate of that of R, but to a smaller scale.* That this should be so we must have

$$\frac{RK}{KN} \text{ a constant.}$$

Or, referring to Fig. 112,

$$\frac{OQ}{QX} \text{ must be constant.}$$

* It must be noted that O R P and P M X are each one piece.

Now R P M Q is always a parallelogram, because its opposite sides are always parallel; therefore P R O is always parallel to M Q, and P M X always parallel to R Q. Therefore the triangles O R Q and Q M X are always similar, and consequently

$$\frac{O\,Q}{Q\,X} = \frac{O\,R}{Q\,M} = \text{a constant.}$$

Hence the point X moves in the same sort of path as O.

Similarly, if the fixed point is situated at I, then the path of J will be a duplicate of that of O, to a smaller scale; or if J is a fixed point then the path of I will be similar to that of O. The same conditions will be fulfilled if the rod M Q were removed to X H, and the rod R Q prolonged to H; in fact, the same will hold wherever M Q is, as long as it is parallel to P R O. It will be noticed that the fixed point and the moving point and its duplicate are always in a straight line.

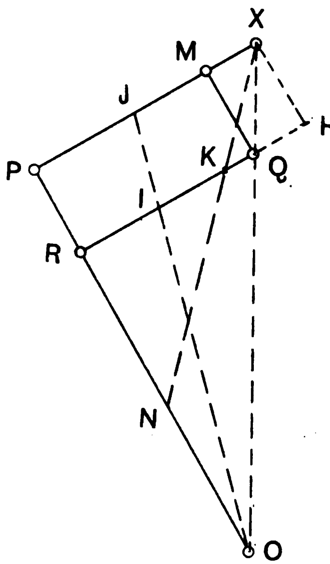

FIG. 11.

If it is desired to alter the rates O Q to Q X, it will be advisable in general to shift the point O along O R P until the required ratio is secured, and then select the point K in R Q, such that it is in the straight line N X. Any desired ratio may be obtained in this way.

An illustration of the pentagraph reducing gear, as applied to an American passenger locomotive, is given in

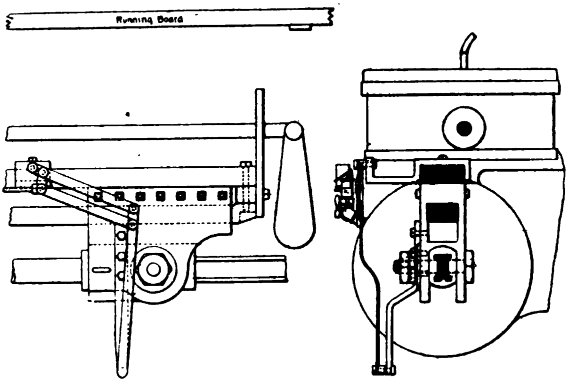

FIG 12.—Badly designed Pentagraph Reducing Gear, attached to a Locomotive.

Fig. 12. The rods, in this instance, were made of 1 in. by $\frac{1}{4}$ in. iron with the joints enlarged to $\frac{1}{2}$ in. thickness.

On account of the vibration caused by the running of the locomotive, and also on account of the want of stiffness of the reducing gear, together with the high speed of reciprocation, it tended to swing sideways, especially at high speed, and was replaced in this particular instance by the gear shown in Fig. 13. It should have been made of stiffer rods, with larger bearing surfaces; but even then the gear as used would be likely to be unstable. Here there is a pendulum rod $2\frac{1}{2}$ in. by $\frac{1}{2}$ in., with a pair of slots A and B in which work a pair of corresponding pins, one fixed to the crosshead and the other to the sliding rod C. The pendulum rod is centred at P to a bracket attached rigidly to the running board and to the top motion bar. A guide bracket for the rod C is also fixed to this vertical bracket.

The reducing gear is perfect so long as there is no slack between the pins and the sides of the slots.

It will be noticed on referring to Fig. 13 that the drum cord is very short and driven by a small collar on the sliding rod C. A detail of this is given in Fig. 14.

The collar has a short arm cast upon it which is triangular in section and has also cut in it a narrow slot to

hold the drum cord. The latter has fixed upon it in the required position a small block of wood. When it is desired

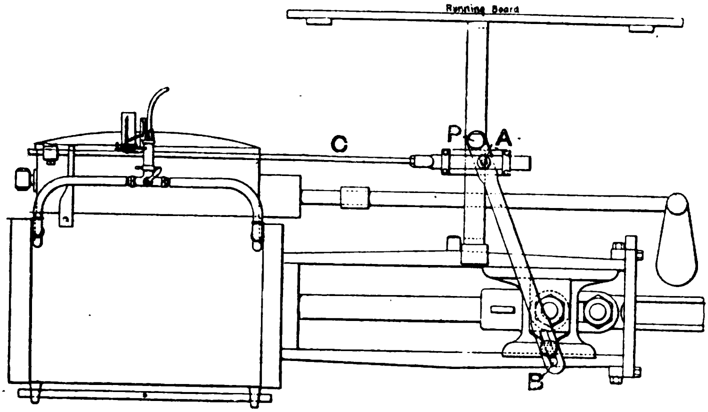

FIG. 13.—Indicator rigged up on American Locomotive.

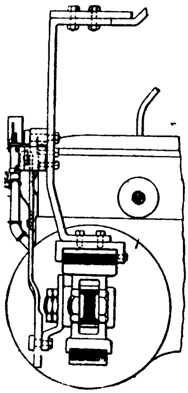

FIG. 14.—End view of Reducing Gear.

to start the paper drum the end of the cord is pulled by hand until the wood block is on the left of the slotted arm and the cord is in the slot. If the cord is now released the wood block will bear upon the slotted arm and the drum will be reciprocated.

To stop the drum the cord is pulled by its loose end and slipped out of the slot latterally. It will be found convenient to make the end of the cord fast to some fixed point, so that the end can be picked up easily.

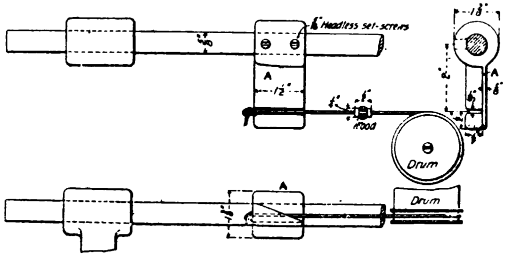

FIG. 15.—Method of Attaching Cord to Reducing Gear.

The reducing gear shown in Fig. 9 is also a perfect one if the indicator cords are not short. The rod A C B oscillates about a pin C in the bracket F, which is itself fixed to the engine frame. The lower end of the oscillating arm contains a pin, which slides in the slotted plate D attached to the crosshead by a couple of set screws. The slotted plate must be packed away from the crosshead, so that the pin B cannot disengage itself through any side motion of the arm. An arrangement for easily hooking on the indicator cord at high speed is shown in the same figure. A piece of steel wire is bent as shown, one end being secured in the end of the arm, and the other bent into and turning loosely in the fixed pin C. The hook can

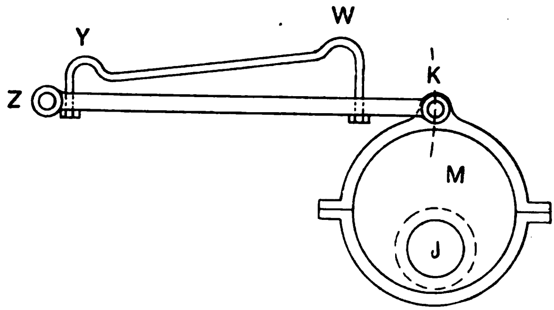

FIG. 16.

be hooked on to the pin end of the steel wire, where the motion is inappreciable, and then slid up the wire until it settles in the loop E. The hook can be easily disengaged by simply taking hold of the cord with one hand and pulling it towards the left, so as to pull out the drum. The hook should then drop out of contact with the oscillating arm.

The bent-wire arrangement was used by Mr. Willans in the manner shown in Fig. 16, for driving the indicator drum when indicating his central valve engines. The eccentric M is attached to the shaft, so that its centre line coincides with that of the crank. The eccentric strap contains a lug K, to which is pinned the rod Z K, oscillating about a fixed pin Z.

The bent wire contains a couple of loops Y and W, the latter being the position of the cord hook when driving the indicator, and the latter when the drum was at rest.

Another arrangement of reducing gear due to Prof. R. H. Smith is shown in Fig. 17. The stroke of the guide block C is S. Two links, B G and G C, are pinned together at G, and coupled to a fixed point at B, and the guide block at C. The point G will always move half as far horizontally as C, because the perpendicular let fall on

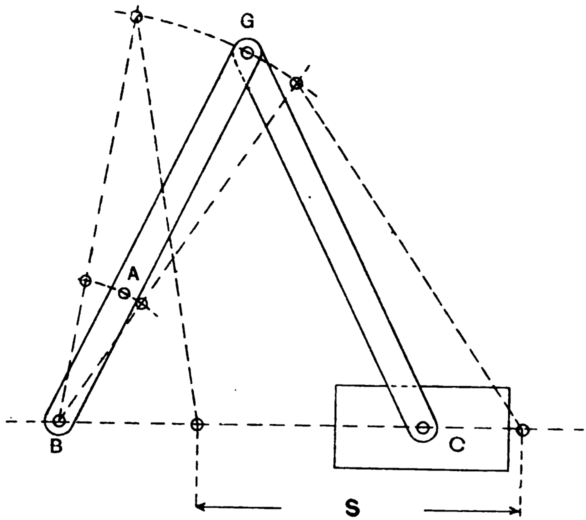

FIG. 17.

B C bisects it. Any point A in B G will have a motion, which is the fraction

$$\frac{BA}{BG}$$

of that of G, hence the horizontal motion of A equals

$$\frac{AB}{2\,BG} \times S,$$

and this being constant the gear is a perfect one, so long as the cord is not short.

Fig. 18 shows a modification of the above gear, with the arm B G, Fig. 117, prolonged to J, Fig. 18. In this case the horizontal motion of J is

$$\frac{EJ}{2\,EU} \times S.$$

Another perfect reducing gear is shown in Fig. 19. A pair of light loose pulleys are connected by a cord or fine

wire. They are supported on pins fixed into brackets attached to the guide bar supports. One of these is arranged for tighting up the cord as shown at Y. The cord is made to partake of the motion of the piston, by a lug or setscrew X. On the sleeve on the pin of the left

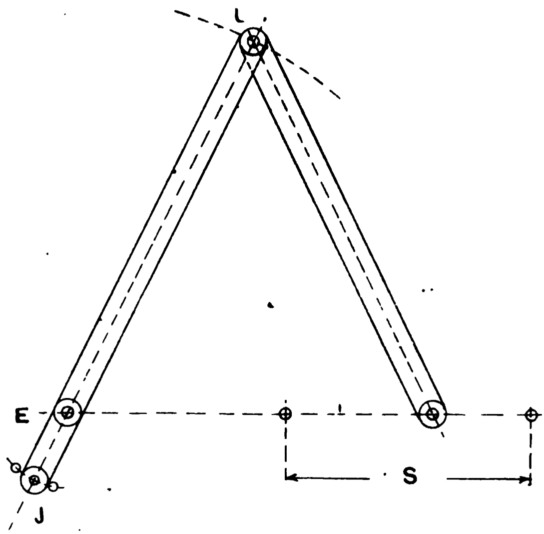

FIG. 18.

pulley is placed a small cord pulley which contains a pin fitting into the boss of the large pulley, compelling them to move together. The indicator cord is wound round the small pulley.

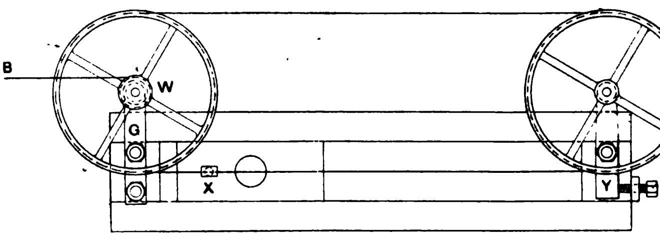

FIG. 19.

A simple reducing gear, which is nearly perfect, is given in Fig. 20. To the crosshead or slipper block is fixed the pin Y, and the link Y Y connects the crosshead

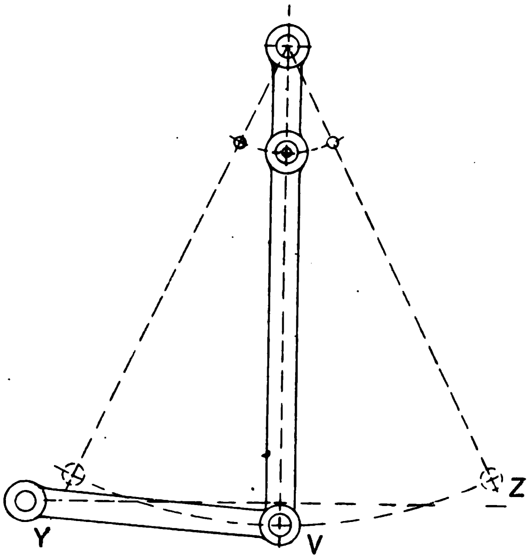

FIG. 20.

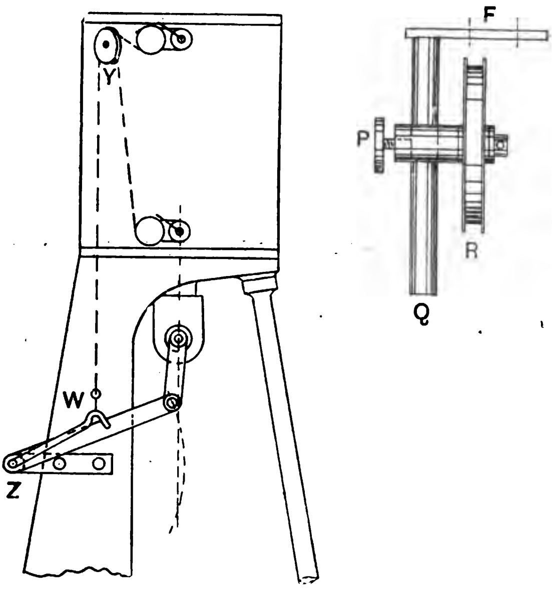

FIG. 21.—A handy Reducing Gear in which the length of cord can be easily and quickly adjusted.

to an oscillating arm. To some convenient point in the latter the indicator cord is attached. If the arc Z V reaches as much above the centre line of motion of the pin Y as it does below it, and the link Y V is not too short, the motion of the drum is almost an exact duplicate of that of the piston, to a smaller scale. This gear is shown fitted to a small vertical engine in Fig. 21. A pair of indicators are used, and the dotted lines represent the indicator cords. The guide pulley Y is shown to a large scale on the right of the diagram, where it is labelled R.

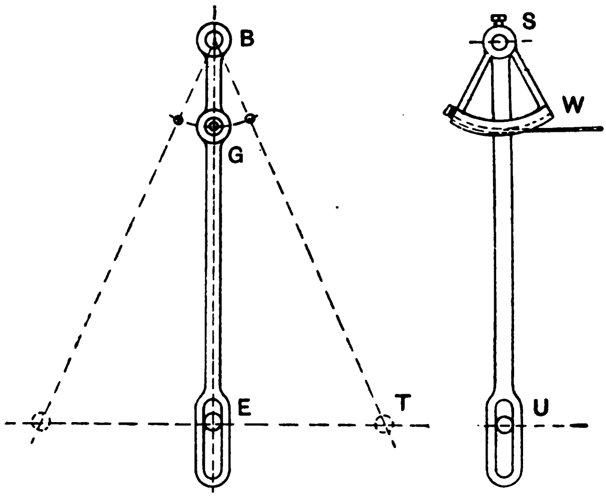

FIG. 22.

Its spindle is adjustable in two directions, which is extremely convenient, being secured on the pillar Q by the setscrew P. The foot F of the pillar may be secured by the cylinder cover nuts. This arrangement will be found convenient for adjusting the length of the cord. The bent wire Z W is shown on the side of the oscillating arm, and the author has found it an extremely handy device.

Of the remaining gears, that shown on the left of Fig. 22 is the most inaccurate. That in Fig. 23 is seldom used, though it may sometimes be found convenient if the cord is long, when it is almost exactly perfect for reducing motion. It has been used by Professor Thurston.

The author is of an opinion that if the reducing gear is reasonably designed, the distorting effect on the mean pres-

sure is inappreciable. He has drawn to scale in Fig. 24 the true diagram and the corresponding diagram as distorted by the reducing gear in Fig. 22, which is about the most inac-

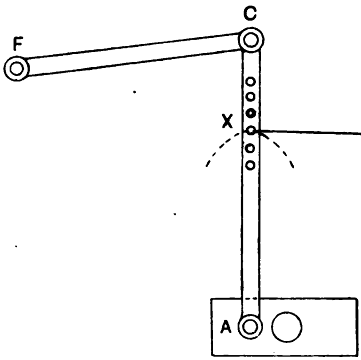

FIG. 23.

curate, and he found that the mean effective pressure of the true diagram was exactly the same as that of the distorted diagram. The reducing gear in this case was quite

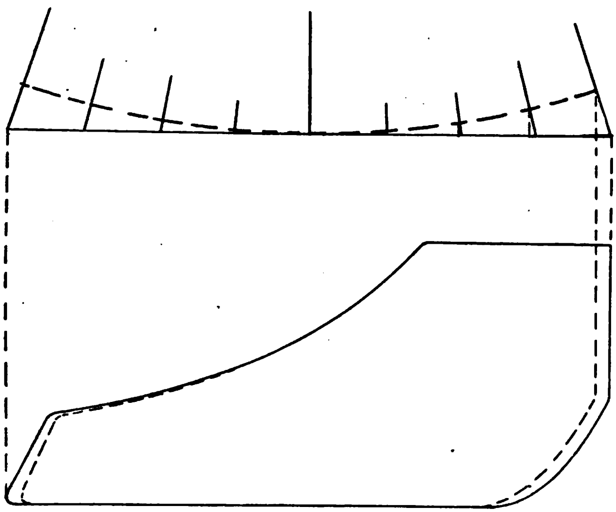

FIG. 24.—Dotted line gives distorted diagram.

as inaccurate as any used in practice, the angle of oscillation E B T on each side of the centre line being about 20 deg. The distorted diagram was 5 per cent shorter than

the true diagram, the distortion occurring principally at the ends of the stroke and then over about 20 per cent of it. This distortion would affect the ratio of expansion as measured on the diagram, but as the point of cut-off is often far from conspicuous, the error introduced would be small.

FIG. 25.—The Tabor Indicator.

Probably any reducing gear described above properly designed would not distort the diagram, so much that the error in the mean effective pressure would be greater than the probable error common to the measurement of the diagram.

CHAPTER III.

OTHER INDICATORS.

The Tabor Indicator, shown in perspective in Fig. 25, and in section Fig. 26, has a pencil mechanism precisely the same in principle as the Thompson indicator, previously described. The controlling link of the latter gear is replaced in the Tabor instrument by a curved slot, in which runs a small roller, the roller turning freely on a pin fixed in the pencil arm. This reduces the weight of the moving parts slightly, and, consequently, the oscillation

at high speeds, but the difference is hardly appreciable. The construction is somewhat similar to the instrument previously described, the chief difference being the double helical spiral spring instead of the single spring. The

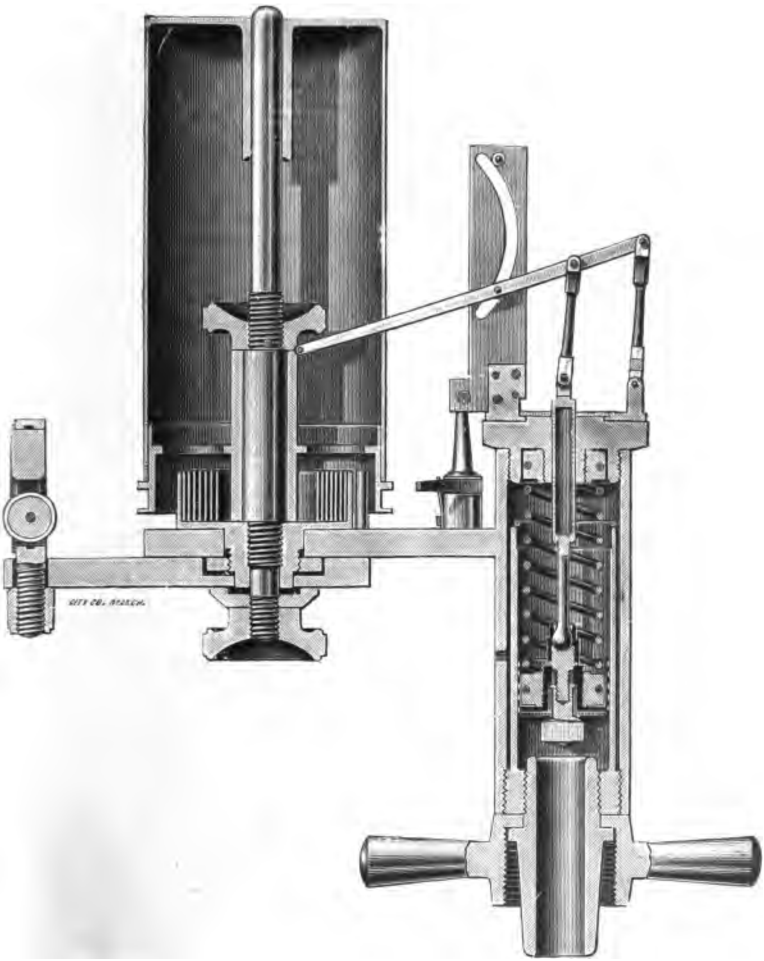

FIG. 26.—Section of the Tabor Indicator.

movement of the pencil is five times that of the piston, thus enabling a taller diagram to be drawn.

The piston rod terminates in a ball and socket joint, to eliminate the possibility of the piston rod grinding in the

cylinder cover, when excessive friction would cause a zigzag outline to the diagram. The piston rod is made hollow for part of its length for the sake of lightness. The piston is also made very light and is grooved on its outer surface. These grooves become filled with condensed steam, and thus in a measure act the parts of packing and lubricant. There is also a stop inside the cylinder to prevent too great a compression of the spring, thus protecting it from destruction if put into unskilled hands.

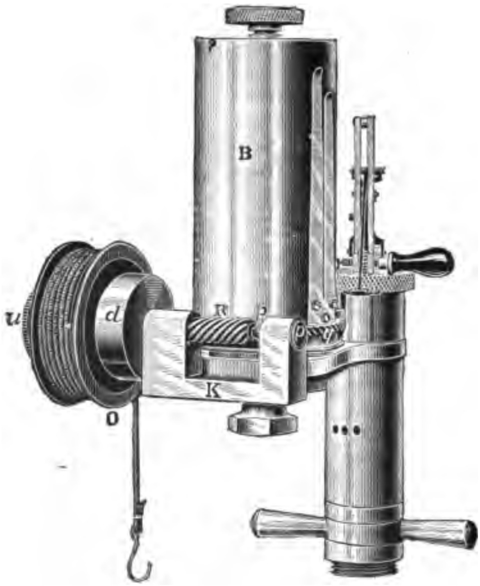

Fig. 27.—Special Reducing Gear Attachment to the Tabor Indicator.

Messrs. Hartley and Co., of Manchester, who sell this instrument in this country, keep the following springs in stock :—

8, 10, 12, 16, 20, 24, 30, 32, 40, 48, 50, 60, 64, 80, 100, 120, 150, 200.

These numbers represent the number of pounds pressure per square inch required to make the pencil move 1 in.

A spring is best selected according to the following rule :—

$$\text{Scale of spring} = \frac{\text{Maximum pressure in cylinder by gauge}}{3{\cdot}1 - \dfrac{\text{revolutions per minute}}{300}}$$

If the scale so found should come between two given above, take the higher of the two.

It will also be found advantageous to reduce the stroke of the paper drum with an increase in speed, and the proper length of diagram with this instrument can be found by the formula

$$(1000 - \text{revolutions per min.})^2 = 160{,}000\,(l - \cdot 5)$$

where l = length of diagram in inches.

A reducing gear which has recently been arranged for some of these instruments is shown in Fig. 27. It is

FIG. 28.—Electrically-operated Tabor Indicator.

simply a species of spiral wheels in gear, and a separate cord pulley with coiled spring. This is said to give good results, but the writer has not used it.

Another attachment to the same instrument is an arrangement of electro-magnets for turning the pencil against the paper. It is shown in Fig. 28. A pair of electro-magnets are fixed to the cylinder casing. An

armature fixed to the swivelling collar carrying the pencil mechanism is attracted by the magnets when current is switched on. With this arrangement, any number of diagrams can be taken simultaneously on different cylinders by one person.

The Crosby Indicator is much used in this country and has a deservedly high reputation. An elevation of the instrument is shown in Fig. 29 and a section in Fig. 30. The pencil mechanism is extremely light, and consequently the effect of inertia is very small. With the exception of the

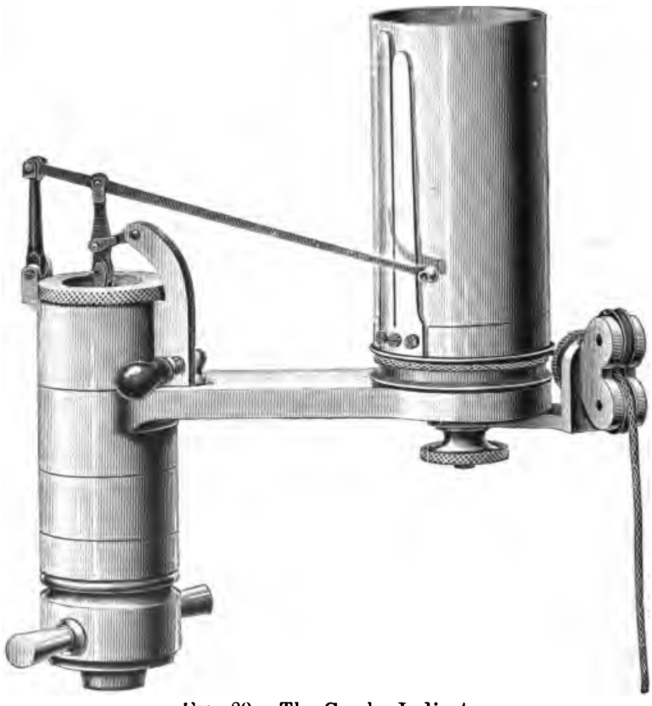

FIG. 29.--The Crosby Indicator.

controlling link, it is the same as in the indicators just described. The piston is of steel, and very light. The piston rod is hollow and is screwed tightly into a split boss on the piston. Into the hollow rod is screwed the shank of the knuckle joint 12 (Fig. 30). The spring is a double helix, threaded into a head-piece and fixed to it, the latter being screwed on to a boss on the cylinder cover.

The lower end of the spring has threaded upon and fixed to it a spherical steel bead, upon which all the piston pressure comes. The bead bears upon the end of the piston rod, and is held in position by the little set-screw 9. The straight piece on which the bead is threaded fits into a pair of slots in the piston boss.

Great care must be exercised in changing springs. To remove an existing spring unscrew the cylinder cover 2, and after removing the piston, &c., from the cylinder, unscrew the spring from the cylinder head. It will be found that before the spring can be entirely removed from the cylinder head the inner piston rod must be unscrewed by continuing to turn the piston and spring.

Next slack back (but not remove) the set-screw 9 in the under side of the piston, and then insert the hollow piston rod into the small box-spanner which accompanies each instrument, and unscrew the piston from the rod. The spring can then be removed and another one put in its place; this is done by reversing the order of the operations above described. Then insert the head into the piston boss, and screw home the piston rod. Next tighten up the set-screw 9. *This last operation must never precede that of screwing home the rod,* or there is every probability of the piston being attached to the spring with its axis slightly out of truth, causing undue friction and a diagram which does not represent the action of the steam in the cylinder. The inner piston rod is next entered in the hollow rod. The spring head-piece can be screwed on the cover boss at once, or the inner piston rod can be entered some distance first, depending upon the range of pressures for which the instrument is about to be used. If the spring is screwed home just after the piston rod has been entered, then the atmospheric line will be near the top of the drum, as shown in Fig. 30, but if the inner rod is screwed some distance in, before the cover boss enters the spring head piece, then the atmospheric line will be low down, as shown in Fig. 29. The former is for the low-pressure cylinder of a compound engine, and the latter for high-pressure cylinders. The pencil moves vertically, through six times the stroke of the piston. The pencil is of brass, and makes a fine line on the paper, but it often becomes polished by rubbing against the surface of the paper, and then the outline of the diagram is almost indiscernible. The point may then be carefully cleaned with a small bit of not too

fine emery cloth,* but this should only be done by an experienced person, as the point may easily be rounded off and rendered useless. The drum, which is partly in section in Fig. 30, is controlled by a helical spring, the tension of which can be regulated by lifting the milled nut, at the head of the spring, and turning it until the desired tension is acquired, and then replace it on the square shank of the drum spindle 28.

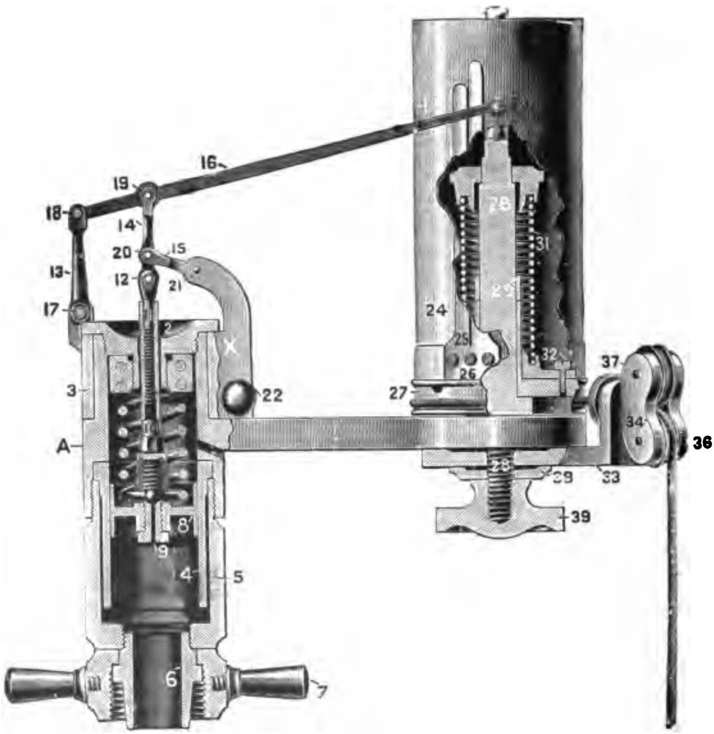

FIG. 30.—The Crosby Indicator.

A small hole is shown under the arm 1 to drain away any leakage from the top of the piston.

The late Mr. Willans was unstinting in his praise of this beautiful instrument, and it certainly does give diagrams at high speeds with very little oscillation.

The piston being of steel necessitates scrupulous cleaning after use, or the instrument may be spoiled by the rusting of the piston.

* A smooth cut file will do as well, the object being to roughen the point so as to give it a tooth.

Altogether this indicator should only be used by experienced persons, and should never be placed in the hands of junior students.

The *Simplex* Indicator is the work of Messrs. Elliott Brothers, of St. Martin's Lane, W.C., and differs from those

FIG. 31.—The Simplex Indicator.

previously described, in having the spring altogether outside of the cylinder. It is of the sugar-tongs shape, and its cylindrical ends slide transversely into corresponding

sockets, one of which is at the end of the piston rod, and the other in the frame of the instrument. A perspective elevation of the complete instrument is shown in Fig. 31, and another of the piston, pencil gear, swivel frame, and spring in Fig. 32.

The paper drum is very similar to those of other indicators, and has a few ratchet teeth just above the cord pulley, into which a small detent can be engaged at will by pushing over the little projecting head on the side of the cylinder. This allows of the drum being stopped without detaching the cord, but such is only used in cases where the detaching of the cord from the reducing gear

FIG. 32.—Pencil Gear Simplex Indicator.

may be somewhat difficult. The pencil gear is of the pentagraph type, and has been used for some time in America on the "Perfection" indicator. It is very compact, and works well, but when cleaning the instrument care should be taken that the links do not become displaced to such a position that the piston rod becomes jammed. In case this happens, take hold of the topmost joint, Fig. 32, and pull it lightly towards the left. This will return the gear to the working position.

There need be no fear of damaging a light spring by excessive pressure, as the end of the piston rod comes against the top of the frame and is prevented from moving further. The convenience of this will be appreciated in obtaining the diagram of the suction stroke in an internal combustion engine.

There is a small groove in the end of the top limb of the spring, into which fits a small peg to prevent the spring from shaking out of its sockets. It would be better to rigidly fix the ends of the spring into their respective sockets, as a small amount of wear will cause looseness in the sockets and result in a certain amount of back-lash.

The author has used one of the larger sizes of this indicator and found it very convenient for gas engine work. The distance of the spring from the steam or gas, as the case may be, allows it to remain at the temperature of the atmosphere, and therefore there is no corresponding correction to be made on that score; while the spring can be changed without removing any part of the apparatus and without the least difficulty.

The swivel frame can only be replaced in one position, when a small dowel pin fits into a corresponding hole in the frame. It is held secure by the milled nut near the top of the frame. The tension of the drum spring can be easily adjusted by means of the milled head at its base.

Type A (large size) gives diagrams 3 in. high, and about 5 in. long for use up to 250 revolutions per minute, while type B gives diagrams 1½ in. in height for use up to 500 revolutions per minute. The following are the instructions supplied with each instrument:—

SIMPLEX INDICATOR INSTRUCTIONS.

Spring.—To fit the spring in position, raise the lever arm until the grooved channels are in such a position as will admit of the ends of the spring freely passing in, observing that the notched end should be uppermost, the notch engaging a spring pin to prevent the possibility of the spring slipping out sideways.

Pencil.—Either a lead or wire pencil can be used. The carrier for the lead being tapped, it makes its own thread on the lead as it is screwed in; and screwed fittings are also supplied with the metal points, thus in each case allowing of fine adjustment on the paper.

Paper Drum.—The paper drum is easily removable by simply pulling it off its fitting.

The spring controlling the motion of the paper drum can be varied in torsion by changing the position of the milled head at bottom of drum stage; small holes will be found in the top of spring box for lubricating purposes.

Cleaning.—Remove the pressure spring, unscrew the broad milled ring, and the piston with the lever motion can be removed bodily.

In replacing, the steel sleeve connection of piston rod must not be screwed home by about half a turn, its correct

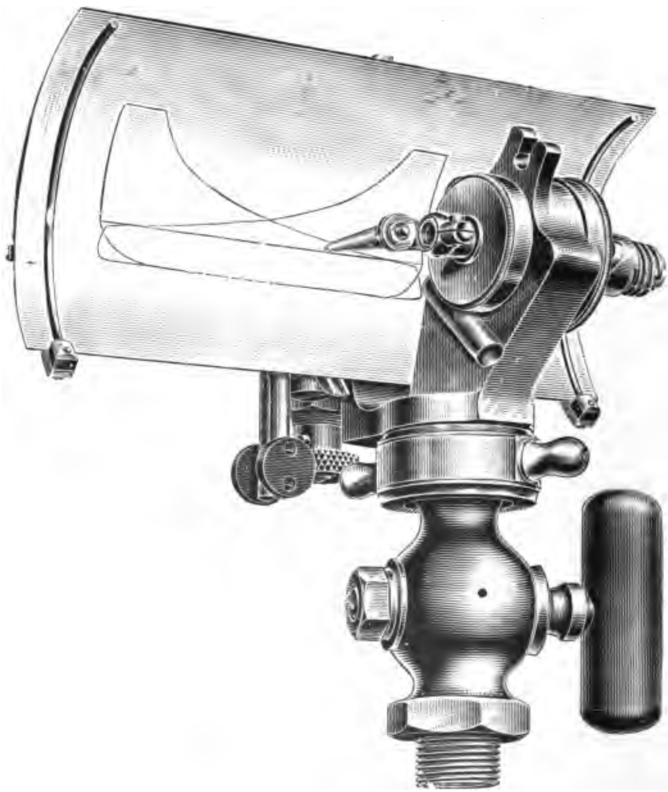

Fig. 33.—The Wayne Indicator.

position is determined by the steady pin engaging in the hole in the top flange. The broad milled ring will then securely fix all in position.

The springs used with both instruments are 10, 15, 20, 25, 30, 35, 40, 45, 50, 55, 60, 70, 80, 90, and 100. With the large indicator a spring may be selected as follows: From 90 lb. per sq. in. upwards:

$$\text{Number of spring} = \frac{\text{maximum pressure}}{3}$$

From 75 lb. per sq. in. downwards:

$$\text{Number of spring} = \frac{\text{maximum pressure}}{3} + 5.$$

With the small instrument, and from 45 lb. per sq. in. upwards, the

$$\text{Number of spring} = \frac{2}{3}\,\text{maximum pressure}.$$

And below 40 lb. per sq. in.:

$$\text{Number of spring} = \frac{2}{3}\,\text{maximum pressure} + 12.$$

The Wayne Indicator is also manufactured by Messrs. Elliott Brothers, and is altogether novel in its design. It is shown in perspective in Fig. 33. The motion of the indicator piston is rotary instead of linear, and the diagram is

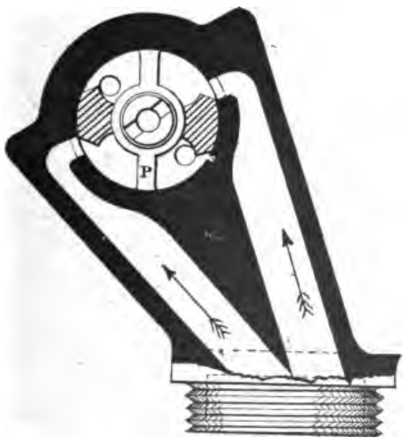

FIG. 34.

drawn upon a card clamped to a concave aluminium frame, whose curvature is that of a circular arc. This frame is made to reciprocate in the direction of the length of the diagram by a cord and return spring, while it slides in a

bed attached to the body of the instrument. The pencil is a small soft brass or hard lead wire supported inside a hollow rod attached to the piston, and pressed lightly against the paper by a weak spiral spring. In the figure it is shown drawing the atmospheric line. The indicator cock, which is of the common type, supports the indicator in the usual way. The body of the indicator is shown diagrammatically in section in Fig. 34. The piston P is a couple of wings, solid, with a hollow piston rod about the axis of which it rotates. Steam or gas enters the indicator through the two channels in the direction of the arrows, and fills the space between the piston wings and the abutments shown cross-hatched in the figure. These latter are rigidly attached to the body of the instrument, and therefore remain stationary, but the pressure of steam on the piston wings turns them round the piston rod axis. Any steam that may leak past the piston finds an easy exit into the atmosphere through the small holes in the end of the cylinder.

A front elevation of the piston and rod is shown in Fig. 35. The spring is shown in position by the

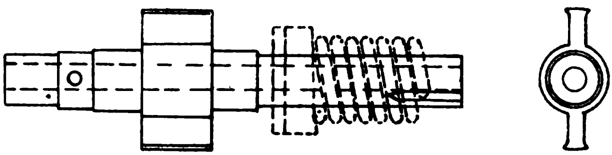

Fig. 35.—Piston and Rod of the Wayne Indicator.

dotted lines. The solid head D (see also Fig. 36) on the end of the spring contains a couple of holes which slide without slack, but quite freely over the two hardened steel pins C C, secured in the body of the indicator. The spring S is slid over the piston rod P, which turns easily in the head piece holding the pins C, and the cross piece B of wire joining the two coils of the spring fits into the V groove at the end of the piston rod P. The hollow nut A, having been previously screwed over the piston rod P, with the straight slot coinciding with the groove in P, is now screwed as much further as it will go, when the spiral grooves in A will jam the spring end well down into the V groove, holding it secure. The slight contraction in the length of the spring as it is twisted is permitted by the head-piece D sliding along the pins C.

The piston is perfectly balanced. It is made of steel, and should, therefore, be very carefully wiped after use to prevent rusting.

The pencil mechanism is the simplest possible, and such as to avoid inertia effects on the diagram. This indicator was selected by Professor Burstall for his researches in gas engine economy for the Institution of Mechanical Engineers. In the place of paper he used sheets of smoked mica, and the tracing point scraped away the soot, making a clear, sharp line. After fixing, this could be used for photographic purposes. The makers supply a special black paper, when necessary, and a special steel point to the pencil, which does not require any sharpening.

The makers recommend a thin clean mineral oil as lubricant, such as gas engine cylinder oil. The following are a few points to be observed in using the instrument.

Adjust pressure of tracer on paper, which can be done by the screw nut at the bottom of cone; if the point is just clear of the paper before bringing the slide forward, it will be right; before putting in the paper bring the paper slide to its forward position, then press the tracing point out to the bottom of the recess in the fixed curved guide. The point should be in the centre of this recess; if it comes on one edge it may tear the paper.

If necessary, adjust pressure with the screwed nut; this nut must not be loose.

The centre wire passes through a small tube or sleeve; the end of the tube is sprung, and should clip the wire tight enough to prevent it slipping, and is intended for a coarse adjustment of the point. This tube plunges freely through the nut, and is held forward by the light spiral spring. The nut is screwed into the large end of the taper tube; by screwing this nut in or out gives a fine adjustment to the point. The wire must not be bent. Before taking a diagram, see that the wire and its tube fittings are free. The wire can be withdrawn for sharpening. All these parts must be quite clean, as dirt will prevent the plunging movement, and tear the paper. The wire and nut may be taken out, and all the parts cleaned in a pot of paraffin.

To place the paper in position, take the paper by the lower corners with each hand, keeping the paper taut, and pass it down the spring clips and parallel with the slide; if working at high speeds, give a little rubbing pressure on

the curved springs, which will press the paper in the recess, and make it impossible to slip.

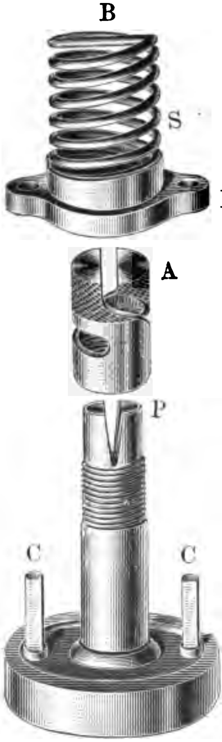

FIG. 36.

When using the white paper, the brass point soon gets polished, and increase of pressure will not make it mark; the point should be sharpened and the polished surface taken off.

If steel wire has been used in the tracer this should be cleaned and dried.

Always use the hard steel point for black paper, which gives a very fine line.

When placing the spring in the instrument, first see that the cross slot on the cap A (Fig. 36) corresponds with the V slot in the piston rod P; then place the cross wire B, of the spring S, in the slot, and turn the piston rod to bring the holes in the spring plate D to the two steel studs C (the spring plate D should slide on these studs freely, but without shake); then press the spring on, and turn the cap A on the piston rod; the spiral grooves will then press the cross wire of the spring in the V slot (the handle of the screw-driver is arranged for turning this cap).

When using the strong springs see that the cap is not screwed too far on the piston rod P, or the cross-wire will fit the V before the spiral grooves can engage the wire; and when fine springs are used, unless the cap is screwed on far enough, the cross wire may come to the end of the spiral grooves instead of being pressed down in the V; the tapped end of the cap A should be kept sprung in so that it will fit the piston rod tightly, but evenly.

For very high speeds special springs should be used.

When ready to take a diagram bring the paper slide to its forward position. Care must be taken that the proper torsion is given to the drum spring for high speeds by turning the milled nut at the bottom of the drum spindle.

Messrs. Elliott make a special lining attachment by

which an accurate diagram may be taken at any speed. It is shown in Fig. 37. Fitted to the crown of the cylinder is a worm, which can be turned by the small handle shown in the figure. A peg piercing the piston rod is moved through a small angle by turning the worm, at the same time it has a certain amount of backlash, so that the pencil can move through a small angle when the worm is stationary. While steam is turned on to the indicator and the

Fig. 37.—The Wayne Indicator, with Special Lining Attachment for very High Speeds.

diagram is being taken, the small handle is turned and the pencil will trace out the horizontal lines, together with small portions of the diagram outline during every revolution. In this way the diagram is built up step by step, and without any trace of irregularities due to inertia. This lining arrangement can be detached, or replaced in a few

seconds, and has been used successfully up to 1,000 revolutions per minute.

The springs used with this indicator range from 5 lb. to 100 lb. for each inch of height by differences of 5. The following are also made: 6, 8, 12, 16, 24, 36, 46, 56, 66, 76, 86, and 96.

In selecting a spring the makers recommend above number 30,

$$\text{Number of spring} = \frac{\text{maximum pressure}}{2 \cdot 5}$$

below 30

$$\text{Number of spring} = \frac{\text{maximum pressure}}{2 \cdot 5} + 5.$$

Little's Integrating Indicator.—This instrument is the invention of Messrs. W. G. and C. W. Little, of Bexley,

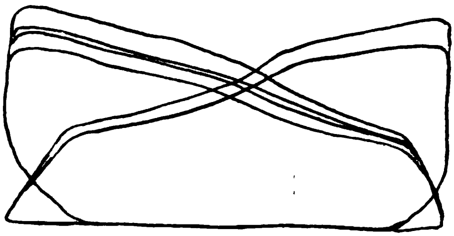

FIG. 38.—Load Constant, Throttling Governor.

in Kent, and is manufactured by the former. It records the work done on the piston of an engine during any required interval of time, and this obviates the necessity of taking a number of diagrams so as to obtain a fair

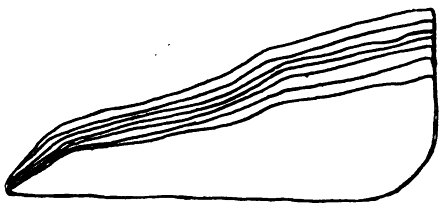

FIG. 39.—Load Variable, Throttling Governor.

average over the given interval. Some idea of the possible variation in the work done during consecutive strokes may be gathered from the following:—

Fig. 38 is a diagram taken from an engine running at 353 revolutions per minute under a *constant load*, applied by means of a rope brake, the speed being regulated by a throttling governor.

Fig. 39 shows the variation of work done under a variable load, the diagram being taken from the same engine.

Fig. 40 shows a card taken from a Corliss engine.

It is at once evident that a few diagrams taken periodically cannot give so true an average result as an instrument which records the work done during every stroke over the same interval of time. But it will be

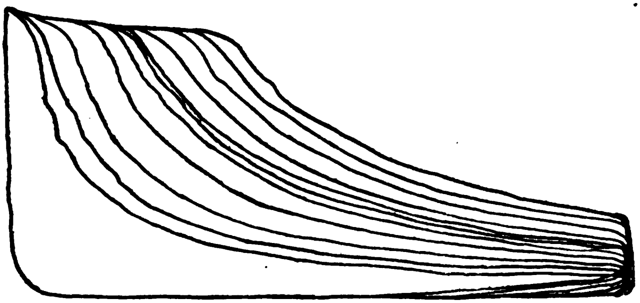

FIG. 40.—Load Variable (Corliss Engine).

shown that the results obtained by the two dissimilar instruments are themselves not very dissimilar, which indicates that whatever may be the faults of the continuous indicator, it is not much less accurate than the older form of instrument.

Comparison with Crosby Indicator.—At various times this indicator has been compared with a Crosby instrument, fixed with it on the same end of the cylinder, with the following results.

Speed. Revolutions per minute.	Kind of Load.	Crosby Indicator M.E.P. lbs. per square in.	Continuous Indicator, M.E.P. lbs. per square in.
350	Constant (dynamo)	21·99	21·93
52	Variable (shop)	28·12	28·27
—	Constant (brake)	26·25*	26·16*
—	Variable (shop)	83·35*	87·42*
96	Constant	14·72*	14·87*
97	Constant	15·55*	15·56*
455	Constant	14·24	14·41

* These numbers are the respective indicated horse powers.

Theory of Integrating Indicator.—To better understand the working of the instrument, it may be as well to state at the outset that it is a planimeter and steam engine indicator combined. In the theory of planimeters given on page 23 of the author's work on "Engine and Boiler Testing," it is shown that if a rod has attached to it a roller wheel with its axis parallel to the rod, and if one end of the rod be constrained to move along any line, while the other end be made to trace round the outline of any closed figure, the area of that figure was recorded by the amount of roll of the wheel multiplied by the length of the arm. Further, that this was true whatever be the fixed position of the wheel along the rod.

Let the wheel be situated at the end N of the rod P N which slides along the line N U, Fig. 41, then the area of

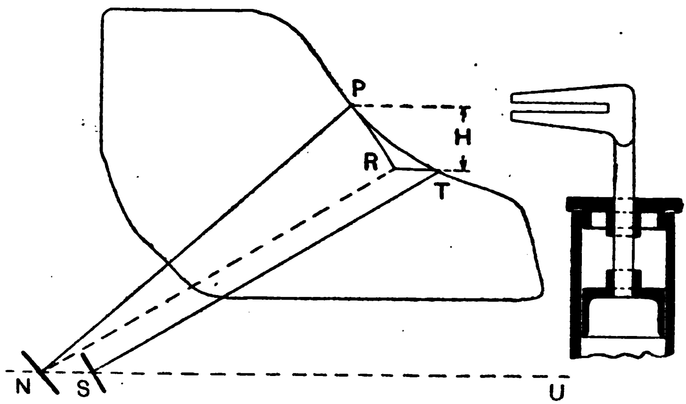

FIG. 41.

the diagram will be given by the roll of the wheel multiplied by the length P N, if the end P be made to trace once round the outline of the diagram.

Consider any *small* piece P T of the outline. The motion of the planimeter rod can be resolved into one of rotation about N into the position N R and then one of translation from N R to S T. When the piece P T is indefinitely small, then P R T coincides with it. The motion of translation is simply the motion of the point of contact of wheel and paper relative to the paper; and it may be produced by the paper remaining stationary and the wheel moving over it; or by the rod N R remaining stationary, and the paper moving parallel to N U from

right to left. This latter is what actually does take place, the paper being replaced by a smooth brass cylindrical surface, which is made to turn upon its axis. The motion of rotation of P N is brought about by the point P being compelled to move through the vertical height H, while the constant length of P N compels P to move in a circular arc round N. The mechanism which moves P through the vertical height H, which must be proportional to so much pressure, is the forked crosshead shown at the right of Fig. 41, it being situated at the end of the indicator piston rod, whose vertical motion represents pressure. Furthermore, both of

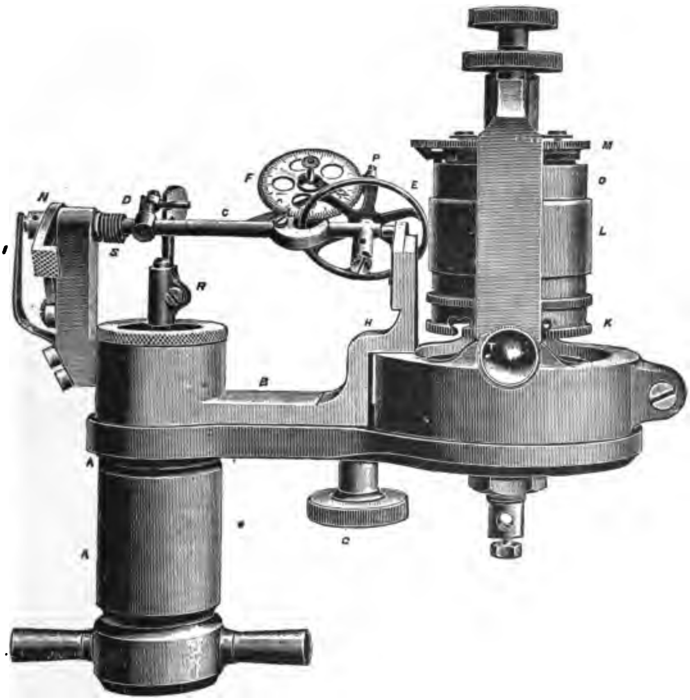

FIG. 42.—Little's Integrating Indicator.

these operations go on simultaneously, and the result, as measured by the rotation of the wheel, is exactly the same as if the point P really traced round the diagram, and P R T coincided with P T.

Description of Integrating Indicator.—The indicator as made is shown in Fig. 42.

The steam cylinder A is of the ordinary Crosby type, and contains a similar piston and spring. The piston-rod end is encircled by a split sleeve R, which is itself the lower end of the fork, shown in this and the preceding figure. The fork, which is hardened, gives motion to the little pin situated in the end of the arm D, which is itself secured on the spindle C. This spindle C is supported at its extremities by the vertical arms of the carriage B which can be turned round the steam cylinder in exactly the same way as the pencil gear of the ordinary indicator. The spindle C carries the spindle of the rolling wheel E, together with the recording wheel F. This latter consists of a pair of wheels on the same axis having respectively 100 and 99 teeth, both of which wheels engage with a worm on the spindle of the rolling wheel E, and consequently 100 turns of the wheel E will be registered by the differential motion of the counting wheels as 1, and hence a great number of revolutions of the wheel E may thus be counted approximately. The rolling wheel E is pressed against the surface of the reciprocating drum L by the light flat spring I at its left-hand extremity, and the constancy of this pressure is necessary for the accurate recording of the work done.

The annulus L is made of hard brass, and can be slid parallel to the axis of the drum if any portion of the surface should become damaged, so that the rolling wheel E shall bear upon a good smooth surface. The drum O is made to reciprocate in the usual manner by a cord round the cord pulley K.

The carriage B, which supports the spindle C, is secured in its proper position by the thumb-screw G.

Comparing this figure with Fig. 41, it will be seen that the axis of the rolling wheel E coincides with the axis N P (Fig. 41), but instead of the forked crosshead controlling the end of the spindle of E (Fig. 42) it controls the end of a parallel arm D, rigidly attached to the axis of E by means of the spindle C, the two results being identical.

A device is shown at M for accurately measuring the stroke of the drum while in motion, and which, by means of a vernier, can be read to ·005 in. It consists of a centre piece and a pair of graduated rings, all concentric with the drum O. A projection on the drum can come in contact with similar projections on the concentric rings, and if these latter are lightly turned by hand until they come against the stop on the drum, the distance between them represents the motion of the drum.

It is found by experiment that any slip due to the friction of the roller on the drum during the forward stroke is compensated by an equal slip during the backward or return stroke.

Rigging up the Indicator.—The instrument is screwed to the cylinder cock by means of the union, and the knot on the end of the cord is inserted into the gap in the lower edge of the drum; the stop T must then be screwed out, taking care, as above directed, to hold the cord tightly to prevent the drum recoiling when the stop is released. Having allowed the tooth on the under edge of the drum to pass the stop, it is again screwed in to hold the drum in the working position until the cord is attached to the actuating lever of the engine.

The length of the cord should be so arranged that the vernier does not come beneath the drum carriage on either side, and if the drum should then touch the stop T it can be removed. The cord, which must not overlap, can now be hooked on, and the stroke of the drum noted. If a low-speed instrument is used, and which is suitable up to 200 revolutions per minute, it is better to make the stroke less than 2 in. at 200 revolutions, say about $1\frac{1}{2}$ in. At 100 revolutions 2 in. may be used, and at lower speeds more in proportion up to 3 in., or even more if it is desired to obtain a larger reading in a shorter time.

When all is ready to commence the indication, turn on the steam cock after blowing off the condensed water, and by means of the rocker N allow the wheel to run in contact with the drum for five minutes to prepare the surface, then take it off again and set the dial to zero. As the time of the run must be noted, it is necessary to do this at the instant of pulling forward the rocker when the integrating gear is set in motion by so doing. It is most convenient to do this by means of a stop watch, which can be started at the same instant that the rocker is pulled over. Another method of starting is to pull over the rocker with the dial set in such a position that it will read *towards* zero, and when this point is reached (*i.e.*, when the small hand fixed to the shaft points to zero) start the watch. To stop the instrument at the end of the reading, the steam cock must be shut off and the time noted. It is, of course, most convenient for purposes of calculation to shut off the steam at the end of a certain minute (*i.e.*, to run for a certain number of minutes). The indicator cannot be stopped by means of the rocker, as the contact with the drum causes it

to spin round when it is taken off, thus destroying the accuracy of the reading.

If indications are to be taken from both ends of the cylinder, the reversing cock, which is generally supplied with small engines, may be turned over at the end of a given number of minutes and the reading noted at that instant. The next reading at the other end of the cylinder will, of course, commence from this point. The dial is numbered so that it may be read working in either direction, the red figures are read to the right, and the black to the left, of zero. The dial is constructed to record 10,000 revolutions of the intergrading wheel. There are two pointers, one on the shaft indicating the units and tens of revolutions of the wheel, and the other on the counter itself, indicating hundreds of revolutions of the wheel—that is, when the dial has made one complete turn the long pointer on the counter will record one division or one revolution of the counter (*i.e.*, 100 revolutions of the wheel). Both pointers must point accurately to zero at the same time, for, if care is not taken to see that this is so, after a prolonged run an error of a hundred divisions or one complete revolution of the dial may be made.

The scale of the spring should never be less than about 100 per cent higher than boiler pressure, and may be used up to double the boiler pressure measured in pounds per square inch according to the speed.

The Indicated Horse Power is computed as follows:—

Let d = diameter of steam engine cylinder in inches.
L = stroke of piston in feet.
S = scale of indicator spring.
σ = stroke of indicator drum in inches.
R = record of counting wheels per minute.
K = constant of instrument.

$$= \cdot 000238 = \frac{\pi}{4} \times \frac{10}{33000}$$

then $$\text{I.H.P.} = \frac{d^2.\ \text{L. S. R. K.}}{\sigma}$$

and $$\text{mean effective pressure} = \frac{10.\ \text{R. S.}}{\sigma \times \text{revs. per min.}}$$

The amount of the stroke of the drum, which corresponds to the length of an ordinary diagram, is measured by means of the graduated device fixed above the drum. When the cord is attached to the actuating lever of the

engine, the drum stroke is measured by bringing the vernier gently round with the finger until it is felt to come in contact with the tooth on the edge of the drum, and the same with the graduated ring, but of course in the opposite direction. When they have both been set by the motion of the drum at the two ends of the stroke, the reading can be taken, and hundredths of an inch read off on the vernier. This device is very accurate, and a little experience is only required to enable the stroke to be quickly noted.

As the drum of the high-speed instruments is only half the diameter of those made for lower speeds, and the diameter of the string pulley between the milled edges is the same on both, the stroke indicator of the high speed instruments registers two inches only, a point on the pulley moving through two inches whilst one on the drum slide only moves through one inch, and the graduations on the ring of the stroke indicator are therefore numbered 1 for a 2-in. length and 2 for a 4-in. In this case, therefore, the tenths actually measure two-tenths in length, and the lines representing them are carried above the scale to distinguish them from the twentieths, which do not exist on the stoke indicator of the low-speed type.

Hints on Maintenance of Indicator.—As a guide to aid the experimenter in properly using and maintaining the continuous indicator in good order, the following hints should be attended to.

Keep the contact surfaces of the wheel E and drum sleeve L free from dust.

Before taking a reading the instrument should be allowed to run with steam on, and the wheel in contact with the drum for a few minutes, especially if the brass slide on the drum has been moved. When the indicator is working properly it will produce a black mark upon the sleeve; and when it is first started, the surfaces being then clean, it is advisable to touch the edge of the wheel with the finger made slightly greasy with black cylinder oil. After this no further oiling is required.

The instrument cannot be taken to pieces without first releasing the small screw in the split sleeve R. Set over the rocker to remove the wheel E from being in contact with the drum, and then remove the screw G under the bracket. The cap can now be unscrewed and the piston removed.

When the piston has been replaced, the screw G must be screwed home to maintain the point of contact of roller and drum in the axis of the spindle C, which should be perpendicular to that of the drum.

This done, it only remains to tighten up the sleeve of the crosshead, which must be set parallel with the crank and resting upon the small shoulder of the piston rod, by which means the proper angular position of the wheel is fixed. If it is desired to change the piston spring, the integrating carriage B must be removed as above-directed, and then, when held firmly in the left hand, the piston and spring can be unscrewed from the cap by holding the piston between the second joints of the first and second fingers of the right hand, in which way a good grip is obtained. The piston rod must then be unscrewed with the hollow key supplied for the purpose, and the spring changed, having first slackened the tightening screw at the lower side of the piston. When the piston rod has been again screwed up it must be passed through the cylinder cap and the sleeve of the crosshead at the same time, the crosshead being held by the fingers of the left hand over the hole in the cap to receive the end of the rod. When all has been set up as before, the screw in the piston must be tightened up with the small screw-driver provided, by passing it through the union of the instrument.

The drum may be taken out as follows: First release the stop, grasping the drum tightly, so that it may not recoil and scrape the tooth against the stop as it is screwed away; then allow the drum to recoil slowly, in order not to injure the teeth of the stroke recorder above, and which will be carried round by the recoil; then release the screw in front of the drum carriage and draw the fitting steadily off; the drum can then be taken out and the spring examined. When the drum is replaced, care must be taken to insert it again into the square of the spring, so that the tooth on the lower edge comes nearest to the stop, for, if this is not done, the tension of the spring will not be the same as before. The spring should be wound up to give a pull on the cord of about 4 lb. rising to 5 lb., for a complete revolution of the drum. A less tension than 4 lb. to 5 lb. is not advisable owing to the pressure of the wheel on the slide. The drum carriage must be carefully fitted down upon the bracket, and to make sure that this is the case, the milled locknut on the top should be released, and the top bearing screwed up slightly. The bracket screw is then clamped

up and the drum set between its bearings with just a small amount of play between them to avoid all chance of abrasion of the points and sockets. It is also necessary to arrange the teeth on the rings of the stroke recorder so that when all three teeth are touching, the arrow on the vernier stands at zero on the scale, and the tooth on the upper edge of the drum is between those on the rings. This position can always be secured by drawing the teeth on the rings together with the vernier tooth to the left of that on the drum. If the lower bearing is for any reason removed, it must be screwed in again to the same height as it was originally, for if it is raised, the tooth on the upper edge of the drum will bear against the stroke recorder and injure the instrument.

The small screws in the plates above and below the stroke recorder are for regulating the friction of the plates upon the rings, which should not be either too free or too tight, in order that the rings should remain where the drum leaves them. The drum spindle and its bearings are hollow throughout in order that they may contain a sufficient supply of oil for long runs. A screw plug is provided at the lower end, which may be removed to allow the oil to run out, and to permit of a wire being passed through. When oiling the spindle the wire should be inserted to drive out the air, so that the lubricant may enter, the plug at the bottom end being then in.

Blaisdell's Continuous Indicator.—In this instrument the cylinder, piston, piston rod, and spring are similar to those of an ordinary indicator.* The top end of the piston rod is screwed into a light semi-circular bracket F. This bracket supports in hardened hollow cone centres P P the spindle S, the ends of which are also hardened. The spindle has secured upon it the friction disc R, by the lock nut N; and also the worm W. This worm engages with a couple of worm wheels G, one of which is graduated and contains 100 teeth, while the other has 99 teeth. A pointer is attached to the back wheel, and indicates on the front wheel graduations. the number of revolutions of the worm and spindle S, together with its friction roller.

Clamped to the arm which supports the paper drum of the ordinary indicator is the bracket *b*, the top end of which supports the spindle *y*, to which is attached a cord pulley shown in section. The spindle *y* is controlled by a

*Fig. 43.

clock spring in the same manner as the paper drum of the ordinary indicator, the forward motion being produced by the cord on the cord pulley, and the return motion by the clock spring. A light and hard steel disc *d* is supported by the spindle *y*, and made to partake of its motion by the set pin *k*, while the small helical spring *c* maintains it in contact with the disc with uniform pressure.

As the disc rotates through a given angle, the friction between the disc and roller compels the roller to rotate as well by an amount depending upon the distance of the

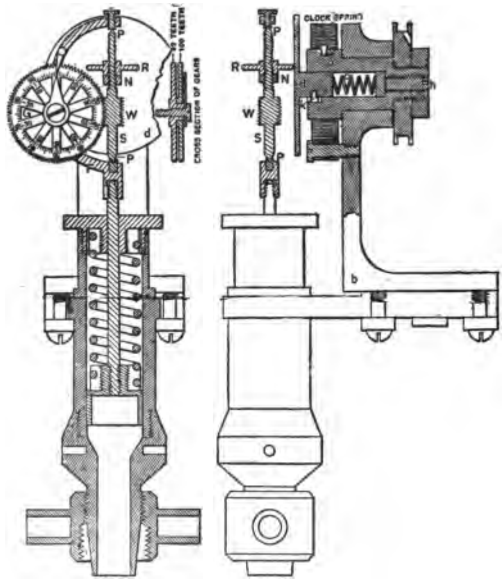

FIG. 43.—Blaisdell's Continuous Indicator.

point of contact from the centre of the disc. The distance of the point of contact from the centre of the disc is the amount the spring is compressed if set centrally when in the zero position, and consequently represents the pressure at the instant in the cylinder. The angle turned through by the roller during any given motion of the disc will be

$$\frac{\text{angle turned through by disc } d \times \text{vertical rise of roller}}{\text{radius of roller.}}$$

As the angle turned through by the disc is proportional to the stroke of the piston of the engine, and the vertical

rise of roller represents the pressure, the angle turned through by the roller R will be proportional to the work done during that interval. During the return stroke the disc moves in the opposite direction, and the work done will be subtracted from that done during the forward stroke, as in the ordinary indicator diagram. The horse power of that end of the cylinder will be the difference of readings of the wheel multiplied by a constant depending on the size of the instrument and the stiffness of the spring, divided by the time in minutes.

It is the intention of the inventor, Mr. B. H. Blaisdell, chief engineer of the Charlestown Power Station of the Boston Elevated Railway Company, to make the indicators complete, and also parts to be added to existing indicators, as in the accompanying illustration. Several tests have shown that this instrument will register the power developed under various conditions with an accuracy equal to that of an ordinary indicator. It is also intended to make instruments with double pistons to indicate simultaneously both ends of the cylinder. This will be a great convenience.

There is a peculiarity about this instrument worth noticing. It is that the roller R need not be set, so that the point of contact is at the centre of the disc when the roller is in its zero position (vertically). For, let θ be the angle turned through by the disc during one stroke of the engine, and, for the sake of simplicity, let the steam pressure be constant during a stroke. If h_1 be the distance of the point of contact from the centre of the disc during the forward stroke, and h_2 during the back stroke, while h is the zero distance, then the reading of the wheel during the forward stroke will be represented by

$$\theta h_1,$$

and during the back stroke by

$$-\theta h_2,$$

so that the net indication at the end of a double stroke will be

$$\theta h_1 - \theta h_2.$$

By subtracting and adding θh to each term we get—

$$\theta (h_1 - h) + \theta h - \{\theta (h_2 - h) + \theta h\},$$

which equals

$$\theta (h_1 - h) - \theta (h_2 - h).$$

The first term is the angle turned through by the disc multiplied by the compression of the indicator spring during the forward stroke, which represents the work done in that stroke, and similarly for the other stroke; hence the roller R may be set anywhere along the spindle S, and will indicate accurately; but, of course, it must not be so set that it may move off the disc altogether.

The pressure of the disc upon the roller can be adjusted by the small screw *h*. This should be as small as possible to avoid excess of friction, and for this reason it would be better to situate the roller R on the lower part of the spindle S, and the worm W upon the upper part, so that the leverage of the pressure between roller and disc, about the indicator cylinder cover shall be as small as possible.

CHAPTER IV.

CALIBRATING INDICATOR SPRINGS.

The Testing of Steam-engine Indicators has been made a study by several eminent engineers, notably Prof. Osborne Reynolds and Mr. Brightmore in this country, and Professors R. C. Carpenter and D. S. Jacobus in America. In 1893 Professor D. S. Jacobus carried out some experiments* to compare the mean effective pressures of simultaneous cards taken with different indicators, and below is the substance of his conclusions:—

1. With correctly-fitted pistons in the indicators, there need be no greater error involved in their use than that which occurs in measuring the hot scales of their springs.

2. A leaky piston is much more reliable than one that is too tight a fit. A piston which will not fall through the cylinder by its own weight may be too tight, and produce errors due to friction. The normal friction of an indicator piston should be so small that there will not be over the thickness of a very fine pencil line between the lines obtained for a rising and falling steam pressure in the tests to determine the hot scale of the springs.

3. The greatest difference found between two indicators, with correctly-fitted pistons for the mean effective pressures of cards taken at one-quarter cut-off, was, as a maximum, ·9 per cent in four indicators tested, and the greatest difference in the mean height of the diagrams for any two indicators was ·005 in. This was for comparative tests, in which the average of 12 or 24 cards was taken. The greatest difference between any single pair of cards taken simultaneously was 2·5 per cent of the mean effective pressure, and the corresponding difference in reading the mean height ·017 in.

4. The greatest difference between any two indicators with correctly fitted pistons for the mean effective pressures of cards having one-fortieth cut off, so that a loop was

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*** Proceedings of the American Society of Mechanical Engineers, vol. xv., page 277.**

formed below the atmospheric line, was ·6 lb., and the greatest difference in the measurement of the mean height was ·01 in., or adding opposite signs, ·017 in. The presence of a loop causes the effect of friction and lost motion to be a maximum, and the above differences are therefore greater than were found in cards having no loop.

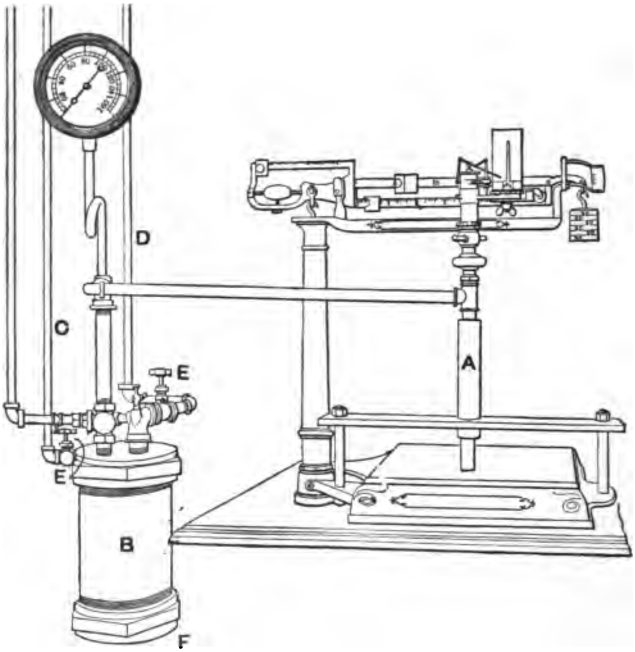


FIG. 44.—Indicator Testing Apparatus.

5. The variation of the weight of the moving parts within the limits now employed by the makers of the standard instruments does not affect the accuracy of the results obtained by the indicators.

Professor Jacobus further states that if the hot scales are determined and applied, there should not be a discrepancy between the results of two indicators of over 2 per cent, but if the nominal scales, as furnished by makers, are used, there may be a difference of 4 per cent.

Professor Carpenter, together with Messrs. Marks and

Barraclough,* have investigated the error in using scales of indicator springs that have been calibrated in the cold state.

Before passing on to the results of the investigation it may be as well to illustrate the apparatus used for the purpose.

Professor Carpenter's Testing Apparatus.—The first piece of apparatus of an extempore nature is shown in Fig. 44. The indicator is attached in the usual way to a pipe A, which is rigidly secured by a bridge to the frame of a small weighing machine. In the lower end fits freely, but steam-tight, a hardened cylindrical plug, which presses directly upon the weighing platform of the machine. This plug is well lubricated, and the pressure upon its upper end is communicated directly to the platform and measured on the weighing scale. The plug can be rotated by hand during the process of weighing to avoid the friction effect.

The tube A is connected to the reservoir B by a horizontal pipe, as shown in the figure, so that the pressure in the reservoir is communicated to the piston of the indicator and to the plug in the lower end of the fixed tube A. The reservoir is connected to a boiler or steam pipe by the pipe C, and to an air reservoir by the pipe D; the third vertical pipe being an outlet for the steam or air in the reservoir B.

Any desired pressure can be maintained in the reservoir B by throttling the inlet with the valve E and the outlet by a similar valve. The pressure in the reservoir B is registered by the pressure gauge shown in Fig. 44.

A more compact form of apparatus, specially designed for the purpose, is shown in Fig. 45, and was made at the New York works of Messrs. Schaffer and Budenberg. The reservoir A is here the body of the apparatus, and supports the indicator to be tested as well as the scale beam. The indicators are attached by cocks to the nipples B. Steam is allowed to flow into the reservoir through the cock F, the supply being regulated by the hand-wheel H. It flows out of the reservoir through the cock G, the throttling being done by the hand-wheel I.

The plug K has a conical end M, which presses upon the stirrup N, which is suspended from the scale beam at O.

* Proceedings of American Society of Mechanical Engineers, page 454, vol. xv.

The top of the cylinder in which K slides is filled with oil to permit of the free working of the plug. A pin P in the plug allows it to be rotated during weighing. By arranging the area of the plug end exactly 1 sq. in. the scale beam indicates the pressure per square inch in the reservoir, and consequently in the indicator.

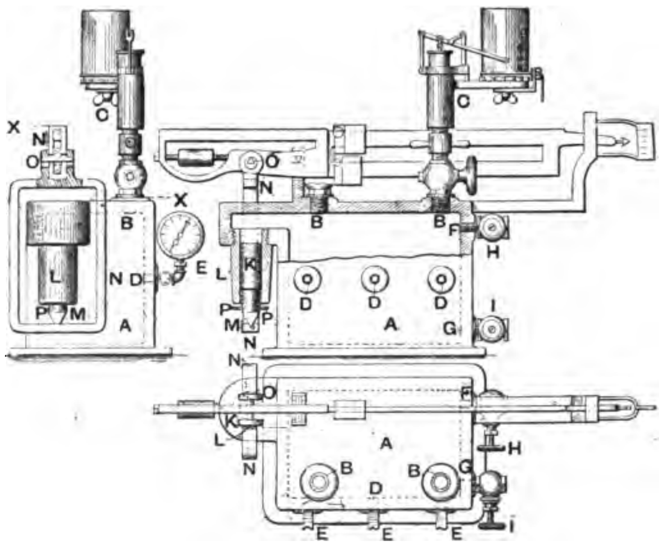


Fig. 45.—Indicator Testing Apparatus.

The method of using the apparatus is as follows:—

The piston of the indicator is oiled, and placed in the indicator cylinder. The weighing poise is set at a given position, to balance a certain load on the plunger K. Steam is then admitted to the reservoir, and the inlet and outlet cocks so adjusted to give the requisite pressure to float the scale beam. The indicator should be quite hot before the test is commenced.

A line is then drawn on the drum paper by the indicator pencil by bringing the pencil up to the paper in the usual manner, and then pulling the drum cord.

This is repeated with successive pressures until the maximum is reached; then, after increasing slightly

beyond this maximum the same is repeated with a falling piston.

Fig. 46 shows the sort of result obtained at an actual test.

A diagram is then plotted on squared paper, with ordinates representing error, and base representing pressures.

It was found in the experiments above mentioned that if the springs are calibrated cold the mean error is about 3·6 per cent. Further, that errors of indicator springs are of such a magnitude that they cannot be neglected, and

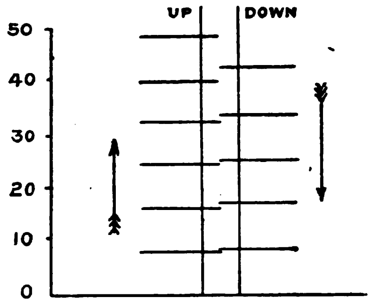


FIG. 46.

these errors are proportional to the pressure, and, consequently, the correction is a percentage to be added to, or substracted from, the mean ordinate.

For ordinary work the weighing-machine portion of the above apparatus could be replaced by an accurately-calibrated standard steel tube pressure gauge, with a large dial, so that small differences of pressure may be indicated by it.

This will be readily recognised when we reflect that, with springs of scale 100, the thickness of anything but a fine pencil line represents a pressure of at least 1 lb. per square inch, which can be easily recognised on a standard gauge such as the one mentioned above.

Quite recently Prof. Jacobus has replaced the weighing-machine part of Prof. Carpenter's apparatus by a weighted plunger.

Professor Jacobus' Testing Apparatus.—A diagram of the apparatus* is shown in Fig. 47. The indicator

*Proceedings of the American Society of Mechanical Engineers, 1898.

is fixed at A, the cock being screwed into the end of a 6-in. pipe B which is about 2 ft. long. This pipe is connected at its lower end to a steam supply pipe L having a stop valve K which can be nicely adjusted. The lower end

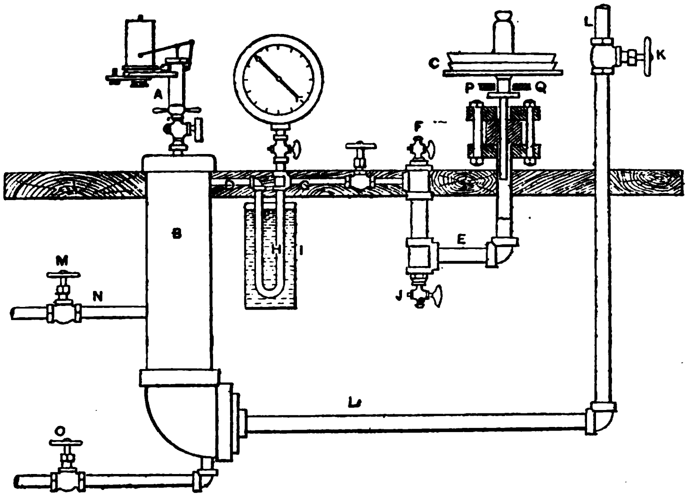


FIG. 47.

of B is also connected to a drain pipe containing the stop valve O. At some intermediate position in the pipe B an outlet pipe N with valve M is provided. Steam being turned on at the valve K passes into the pipe B and out through the pipe N; any desired pressure being maintained in B by suitably regulating the inlet and outlet valves K and M. The pressure in B is measured by the deadweight C pressing upon a piston or plunger of known diameter working in a steel die block. Both the plunger and die block are accurately finished to the diameters ·5 in. and ·5005 in. respectively. So that the effect of friction shall be eliminated, the weights C and their supporting platform, which is rigidly attached to the plunger, are rotated by hand while the actual pressure weighing takes place.

The pipe E is filled with oil so that the plunger is always well lubricated. Glycerine is also found to behave well.

The pet cock F permits any air in the connections to flow out, and the cock J is for getting rid of water in the same manner. The pipe D connects the U piece E to the

pipe B through the water jacketed siphon H, the jacket provides that the siphon H and pipe G is full of water. A pressure gauge shows the pressure at which the instrument is being worked.

When testing a spring, the steam pressure in the pipe B is first raised to the maximum allowed for that spring and the indicator cock is opened and closed a number of times to allow of the indicator being heated to the ordinary working temperature. The valve K is then shut off and M opened for the purpose of taking the atmospheric line. For this purpose the indicator cock should be turned to its proper position. The drum of the indicator is then rotated by hand and a line drawn, the pencil being previously pressed downwards. Immediately afterwards the pencil is pressed upwards and another line drawn. If there is an appreciable difference between the two lines, there is an

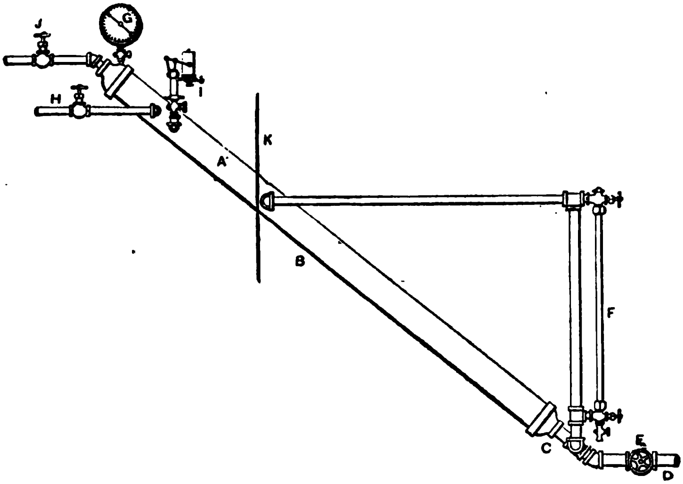


FIG. 48.

excess of friction above the normal amount.

Steam is then turned on and a definite pressure in B obtained, which just balances the plunger and its superimposed weights, the latter being spun round during adjustment.

For pressures above that in a neighbouring boiler, Professor Jacobus used the apparatus in Fig. 48.

An inclined pipe A B C, 4 in. in diameter and 8 ft. long,

is fitted with a water gauge F. The indicator is attached at I and a pressure gauge at G. The pipe H leads to the pressure weighing apparatus as in the last figure, and the valve J is in the escape pipe. The pipe D admits water or steam to the pipe B. The lower part of B is heated with Bunsen flames to any desired temperature, an asbestos sheet K preventing the upper part of the pipe receiving heat from the gas jets.

The pressure in B is regulated by the valve J. The upper end of the pipe A B is well covered with non-conducting cement.

Fig. 49 shows the apparatus fitted up for calibrating springs below the atmospheric pressure. The pressure in

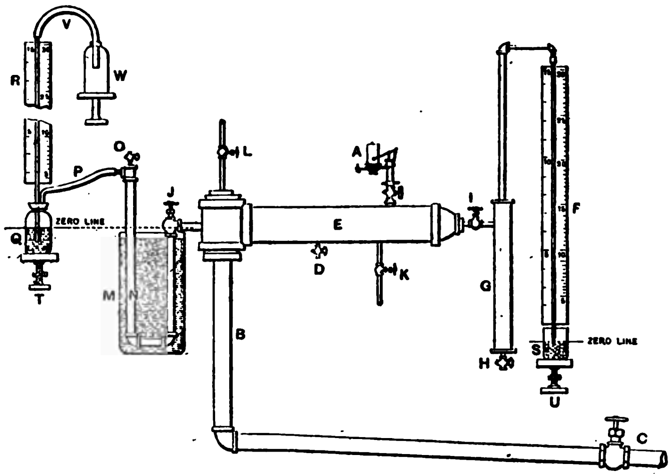


FIG. 49.

E is measured by a mercury column F connected to the pipe E through a condenser G to prevent any steam gaining access to the top of the mercury column and condensing there.

The indicator is attached at A, while the pipe B leads to a vacuum pump, the amount of vacuum obtained depending upon the adjustment of the valve C, and the air cock D. Steam is admitted through the cock K for heating up the indicator.

The apparatus can also be used for small pressures above the atmosphere by the mercury column R on the left. The mercury cups, Q and S, are adjusted by screws T and U beneath them.

CHAPTER V.

OTHER ERRORS IN THE INDICATOR DIAGRAM.

Errors in the Indicator.—In 1885 Prof. Osborne Reynolds contributed a highly-interesting paper to the Proceedings of the Institution of Civil Engineers, vol. lxxxiii., "On the Theory of the Indicator and the Errors in Indicator Diagrams," and with it was read another paper by Mr. A. W. Brightmore, B.Sc., entitled "Experiments on the Steam-engine Indicator."

The discussion which followed these two papers was voluminous and valuable, and should be read by any one interested in indicator work.

In the former paper Professor Reynolds shows that the error in pressure at any instant due to the effect of the inertia of the piston and its attached mechanism could be expressed by an equation which consists of two parts: one relating to a cyclic disturbance whose period was that of one revolution of the engine shaft, and the other relating to a vibratory disturbance whose period was that of the piston and pencil mechanism when disturbed and depended upon the dimensions of the moving parts and the stiffness of the spring.

It was also shown in the paper referred to that the principal effect of the cyclic disturbance was to slightly enlarge the area of the diagram, though with proper precautions this should be inappreciable. The probable error per cent due to this cause would be about

$$\frac{\cdot 00563\ W\ N^2}{A\ E\ S} \text{ where}$$

N = number of revolutions per minute.
A = area of indicator piston in square inches.
E = number of pounds to the inch of spring scale.
S = ratio of pencil motion to that of indicator piston.
W = equivalent weight of piston and pencil mechanism moving in the same manner as the piston does
= $\Sigma\ (w\ r^2)$ where
w = weight of any particular piece of pencil or piston mechanism, and
r = ratio of its motion to that of the piston.

The error due to this cause should never be allowed to exceed 1 per cent. This may be done by selecting a spring of sufficient stiffness.

Oscillation of Pencil.—In the vibratory disturbance the number of complete oscillations of the pencil during one revolution is given by

$$n = 189\,\mathrm{N}\sqrt{\frac{\mathrm{A\,E\,S}}{\mathrm{W}}}$$

where n is the number of oscillations per revolution.

It is stated in the paper that n should not be less than 30 to obtain a fair diagram.

The effect of these oscillations of the pencil is to produce a distorted diagram rather than alter its area to any great extent, though Mr. Brightmore has shown that a slight increase in the area is produced by this phenomenon. Friction tends to destroy the vibration, but though friction may be useful in this respect, it should be eliminated as far as possible, as it tends to produce error in other ways, principally by delaying the motion of the pencil and consequently increasing the area of the diagram. This occurs mainly during expansion and compression. Another effect of the friction spoken of above is to round off some of the sharp corners of the diagram.

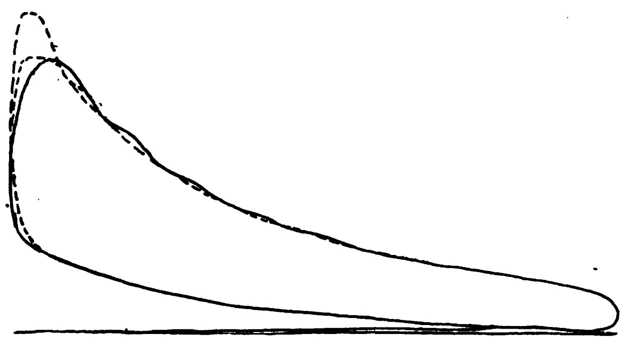


FIG. 50.—Typical Normal Gas Engine Diagram (Lighting Gas) Full Load.

A typical diagram taken with a large Simplex indicator, and a 100 spring attached to a gas engine, is shown in Fig. 50. As diagrams from gas and oil engines are very similar in character it may be taken for our purpose as a fairly representative diagram from an oil engine, when the proper spring is used. The speed was 200 revolutions per minute. More oil engine diagrams will be shown later.

Fig. 51 shows an oil engine diagram taken with a Richards indicator, and a 64 spring at 225 revolutions per

minute. Here the oscillation of the pencil is excessive in amplitude, extends throughout the stroke, and shows well the influence of the heavy pencil mechanism in this indicator. The spring is slightly weak for this work, but would

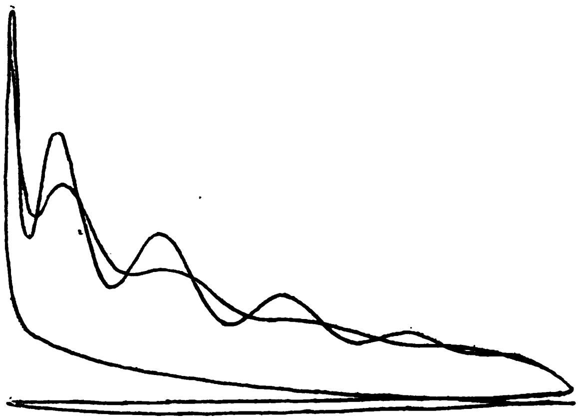


FIG. 51.—Oil Engine Diagram Showing Oscillations of the Pencil of a Richards Indicator with a 64 Spring.

not show so much oscillation if used on a more modern instrument, such as any of those previously described. It is on this account that the Richards instrument has not been mentioned in this work, and one could hardly think

FIG. 52.—Oil Engine Diagram.

of buying it for general or accurate work when other much superior instruments are on the market at a reasonable price.

Another oil engine diagram is shown in Fig. 52, taken at a speed of 235 revolutions per minute with a Richards

indicator. The pressure of the pencil on the paper was probably greater in this than the previous diagram, as the oscillations did not extend throughout the stroke as in Fig. 51.

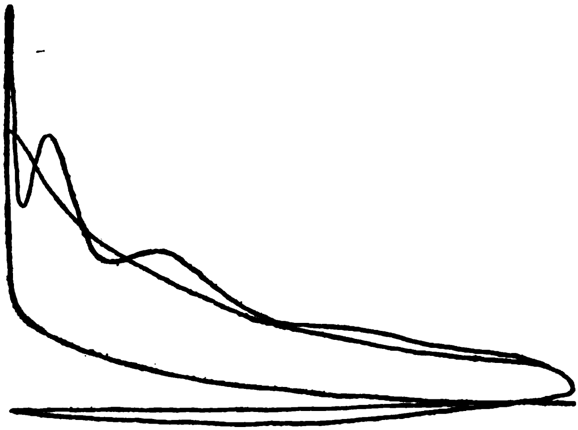


FIG. 53.

Fig. 53 shows much oscillation during one stroke and scarcely any during another. The absence of oscillation in the latter is accounted for by the impulse being slightly less than that during the former stroke and the pencil pressing hard against the paper, friction destroying the oscillations.

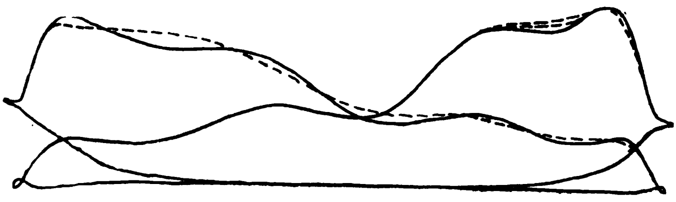


FIG. 54.—Spring too weak (No. 30). Speed, 350 Revolutions per minute.

Fig. 54 gives diagrams taken from a steam engine with a large Simplex indicator and a 30 spring at 350 revolutions per minute, the oscillation being conspicuous.

Of course, so light a spring should not be used for that pressure and speed. Many of the diagrams given in this and other articles were purposely taken with springs of

too small a scale, or with a large slow-speed instrument, when a small high-speed instrument should have been used; the idea being to show what happens under such circumstances. In this way the peculiar results obtained with the highest class of instruments are not due to the instruments themselves, but to the way in which they were handled. There are other features of these diagrams, which will be examined later.

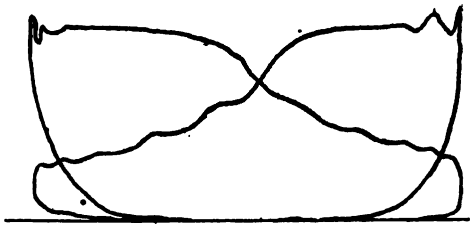


FIG. 55.—**Showing Vibration of Indicator Pencil. Speed, 338 Revolutions per minute.**

Fig. 55, which was taken from a steam engine at a speed of 338 revolutions, shows a small amount of oscillation of the pencil.

Indicator Sticking.—A bad case of indicator sticking is shown in Fig. 56, at a speed of 200 revolutions per minute. It can always be distinguished from oscillation by the form of the zigzag curve. With a sticky piston the

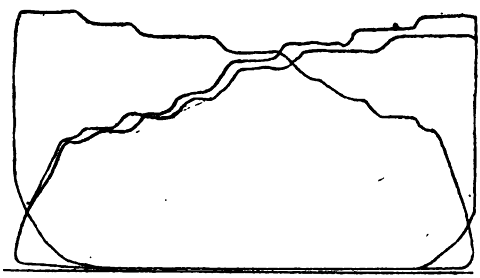


FIG. 56.—**Bad Case of Sticking.**

pencil does not begin to move until after its proper time, and when it does start its inertia sometimes carries it a little too far, where it sticks again. This produces a stepped line, which is partly *horizontal* and partly sloping

at a considerable angle to the horizon. The oscillation curves in previous figures extend nearly as far below the mean position as above it. This does not happen when the piston sticks. Fig. 57, which was taken at the same speed and with the same spring (60) as Fig. 56, shows a slight amount of oscillation, as well as excessive sticking.

The author has seen similar diagrams produced by a Crosby indicator when the piston has been carelessly attached to the piston rod, and was a bit out of truth,

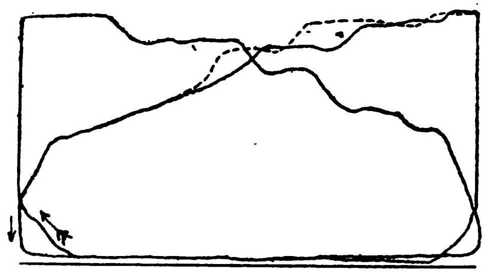


FIG. 57.—**Indicator Piston Sticking, and Slight Oscillation.**

making it grind against the cylinder and the rod against the cylinder head. To prevent this the little screw at the bottom of the piston should always be screwed in *after* the piston rod has been screwed home into the piston-boss.

This is only another instance which shows how necessary it is to exercise great care when using high class and delicate apparatus.

If an indicator shows sticking, at once remove the piston and oil it, after examining to see if the sticking is due to wrong adjustment. A slight amount of leakage past the indicator piston does not materially affect the diagram, while it indicates that the piston does not fit the cylinder too tightly.

Errors due to Paper Drum Inertia.—There remains to be considered the influence of the paper drum and its mechanism on the form of the diagram. This may be considered under three headings, namely: (1) The inertia of the drum, (2) the varying tension of the drum spring, and (3) the effect of drum friction.

The first of these should be made as small as possible by reducing the weight of the drum as far as is consistent with strength and utility; its numerical value for any drum will

vary as the square of the speed of the engine, and its general effect is to lengthen the diagram.

As the effects of (1) and (2) taken together can be made a minimum by the proper selection of dimensions, we shall now calculate its approximate numerical value.

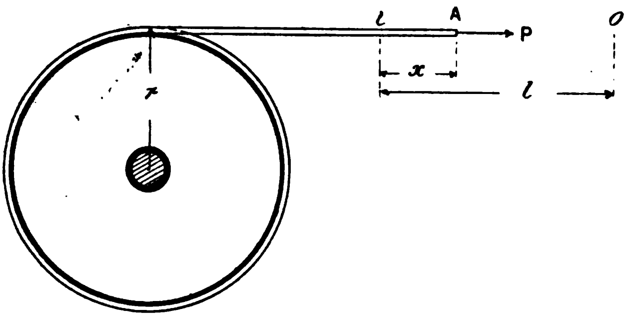


FIG. 58.

In the accompanying Fig. 58 let the paper drum cord pulley have an effective radius of r feet (= radius of pulley + radius of cord). Further, let A be the point of attachment of the cord to the reducing gear, the travel of A being $i\,o$.

Let x = distance of A from i in feet.
l = length of stroke $i\,o$ of A in feet.
L = length of stroke of engine in feet.
P = actual pull on cord in pounds after A has moved x feet from i.
p = initial tension in cord in pounds when A is at i, and engine stationary.
f = that part of the tension in the cord, due to inertia of drum, in pounds.
z = extra tension of cord in pounds beyond initial tension to move A through 1 in. without inertia.
N = number of revolutions of engine shaft per minute.

Then $\quad P = p + 12\,x\,z + f.$

The point A will move in a similar manner to the piston, but to a reduced scale of.

$$\frac{l}{L}$$

The motion of the piston is not quite simple harmonic, but

is approximately so, and will be assumed to be so in this investigation. The actual acceleration of the piston for any angle θ turned through by the engine crank from the centre line of motion, is

$$-\alpha^2 R\left(\cos\theta - \frac{\cos 2\theta}{n}\right)$$

Where α is the angular velocity of the crank in radians per second, R is the length of the crank in feet, and n is the number of cranks in the connecting rod length.

The second term in the bracket is due to the angular motion of the connecting rod, and if this be neglected the acceleration is that due to simple harmonic motion. The factor R cos θ is the horizontal component of the distance moved by the crank from its mid-position, which is the same as the distance of the piston from its middle position (assuming its motion to be simple harmonic). The acceleration of the point A will be that of the piston multiplied by the ratio

$$\frac{l}{L}$$

and equals

$$\alpha^2 \times \text{distance of A from its mid position}$$

$$= \alpha^2\left(\frac{l}{2} - x\right)$$

$$= \left(\frac{2\pi N}{60}\right)^2\left(\frac{l}{2} - x\right)$$

This is the linear acceleration of the point A. Referring to Fig. 55 the linear acceleration of a point in the drum r feet from the axis must equal the angular acceleration × radius r.

Therefore

$$\text{angular acceleration of drum} = \frac{\text{linear acceleration of A}}{r}$$

Now with any body, free to rotate about an axis, we have

$$\left.\begin{array}{r}\text{Moment of turning}\\ \text{force about axis}\end{array}\right\} = \left\{\begin{array}{l}\text{Moment of inertia of body}\\ \text{about axis} \times \text{angular acceleration about axis.}\end{array}\right.$$

Calling I the moment of inertia of the drum, and substituting from above, we have

$$f\,r = \frac{I}{r} \times \left(\frac{2\pi N}{60}\right)^2\left(\frac{l}{2} - x\right)$$

and

$$f = \frac{I\,\pi^2\,N^2}{900\,r^2}\left(\frac{l}{2} - x\right)$$

substituting in the equation

$$P = p + 12\,x\,z + f$$

we have

$$P = p + 12\,x\,z + \frac{I\,\pi^2\,N^2}{900\,r^2}\left(\frac{l}{2} - x\right)$$

This may be constant when

$$12\,z = \frac{\pi^2\,N^2\,I}{900\,r^2}.$$

that is when

$$N = \frac{r}{\pi}\;\frac{\sqrt{10800\,z}}{\sqrt{I}}$$

Hence there is one particular speed of the engine and only one for each drum spring, at which the tension of the cord is constant.

This is represented graphically in Fig. 59. O X is a horizontal base line, and on it are plotted upwards ordinates

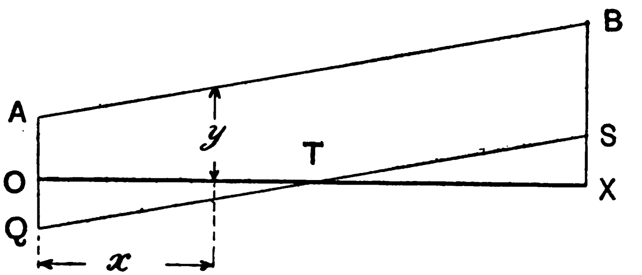


FIG. 59.

representing the tension in the drum cord without inertia, that is when the engine shaft is only just turning round. In this way the line A B is produced, and its equation is

$$y = p + 12\,z\,x.$$

Then are plotted downwards from O X ordinates representing that part of the tension due to inertia, and given by

$$\frac{\pi^2\,I\,N^2}{900\,r^2}\left(\frac{l}{2} - x\right)$$

This gives the line Q S, cutting the base line in T. For the first half of the stroke this tension is superimposed on

that of the spring (because the cord has to give it motion), with the result that at the beginning of the stroke the tension in the cord is represented by

$$A\,O + O\,Q = A\,Q.$$

At the other end of the stroke inertia is assisting the cord, or in other words the slowing down of the drum requires some retarding force to produce it, and that is found in the increasing resistance of the coiling spring; hence the cord will have less to do, with this position of the drum, than as if the engine were running dead slow and the forces of inertia were inappreciable; therefore the tension of the cord at this end of the stroke is given by

$$B\,X - S\,X = B\,S.$$

During the return stroke the spring has to give motion to the drum, and consequently the tension of the cord will

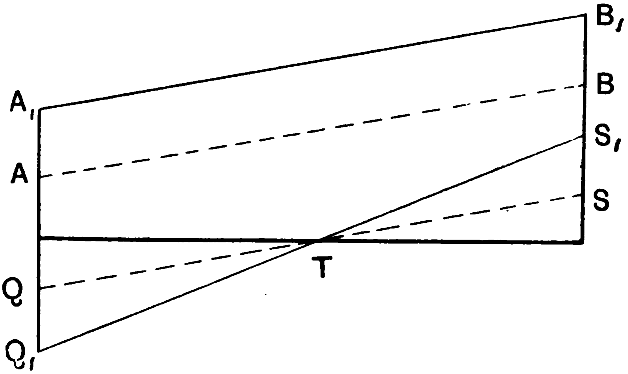


FIG. 60.

be B S at the commencement and A Q at the end. If the speed is such that Q S is parallel to A B, then the tension of the cord is constant throughout the stroke—a most desirable result, as the stretching of the cord would then be constant, and there would be no error due to that cause.

At any other speed the tension cannot be made constant, even by increasing the tension of the spring; for if in Fig. 60 the dotted lines represent the state of affairs with constant tension, the winding up of the spring simply means that the initial tension is increased, and the line A B is moved parallel to itself through a vertical distance representing the increased tension to the position

A B. An increase in speed will rotate the line Q S round T into some position $Q_1 S_1$, which gives an ordinate for the tension greater at one end than the other. Hence there is one particular stiffness of *drum* spring which is most suitable for a particular speed, and no alteration of its tension will render the tension of the *cord* constant. It is, therefore, preferable for very accurate work to have a number of drum springs of different stiffnesses, so that the inclination of the drum spring tension line shall be approximately parallel to the inertia line.

There is one condition which is easily understood from the diagram, and which is extremely undesirable. It is that in which the stiffness of the spring is small while the speed is so great that the inertia line crosses the spring tension line, and the tension in the cord is entirely removed for a certain portion of the stroke, with the probability of the drum considerably over-running the cord and giving a very distorted diagram.

If it is required to use the true acceleration diagram instead of the simple harmonic diagram, the true diagram may be drawn from the dimensions of the engine in the usual manner, and then reduced in scale in the proportion of L to l.

It will be noticed in the expression for the tension of the cord that the maximum force due to inertia is proportional to the length l of the diagram. Hence, if we reduce the length of the diagram we reduce the inertia forces, and consequently we can, by altering the length of

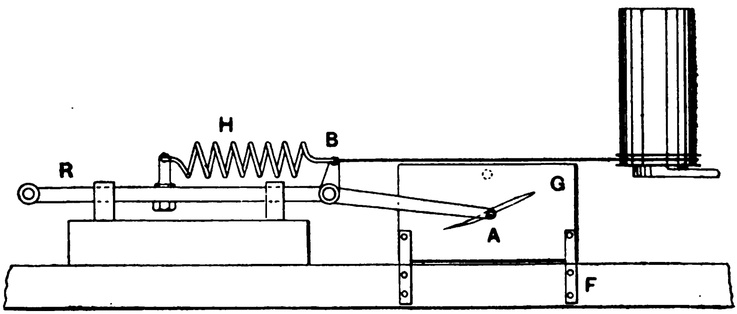


FIG. 61.—Brown's Drum Spring Testing Apparatus.

the diagram, keep the inertia line of previous diagrams parallel with the drum spring tension line. This is recommended by the Tabor indicator makers, who give

instructions in their pamphlets for accomplishing it. (See p. 25.) This also gets over the difficulty of carrying about a number of drum springs, but at the sacrifice of a fair size diagram.

The proper speed or length of diagram for uniform cord tension can be obtained experimentally with the apparatus shown in Fig. 61, which was first used by Mr. Brown, in America. It consists of a rod R, which is made to partake of the same motion as the end of the cord in a reducing gear, by being attached through a connecting rod to a crank pin in some artificial crank, such as the faceplate of a lathe. At the other end of the reciprocating rod is attached a bell-crank lever, the outer end A containing a pencil which can describe a diagram on a card fixed to a small vertical wood frame G, which can be brought into contact with the pencil A.

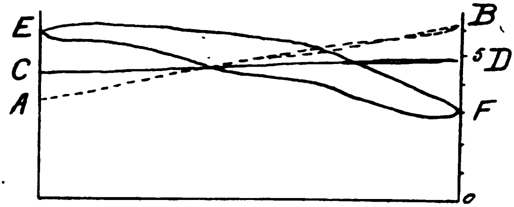


FIG. 62.

The small end B of the bell-crank is connected to a helical spring H, the other end of which is coupled to a projecting arm on the reciprocating rod R.

To the end B of the spring the indicator cord is attached, and its length is the same as that when in use on an engine, the drum being securely fixed to some rigid base; preferably by leaving it intact on the indicator and fixing the latter on a special fitting which is rigidly fixed.

When the reciprocating rod is in motion and the tension of the spring H is uniform, the pencil A will describe a horizontal line; but should the tension of the cord, and consequently of the spring, vary, the pencil will describe a closed curve. Examples of these will be found in Fig. 62. The tensions are plotted vertically on a stroke base. The straight dotted line A B is that described by the pencil when the mechanism is going dead slow, the straight line C D when making 400 revolutions per minute, this being the speed for which the stiffness and length of this drum spring are most suitable; and finally the curve

E F which was drawn at 650 revolutions per minute showing considerable variation in the cord tension.

The effect of the inertia of the drum in stretching the cord at higher speeds is shown in Fig. 63, at the admission and exhaust lines, it being greatest at the admission line of the left diagram, the inertia of the drum making the tension there greater than at the end of the outward stroke on the right.

The diagrams were taken with a large Simplex instrument and a 60 spring at the three speeds mentioned without removing the paper from the drum. The drum spring was not adjusted to give constant tension as it should have been.

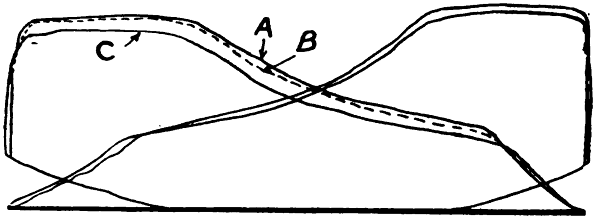


FIG. 63.—Showing Stretching of Cord Due to Higher Speed.
Diagram A, taken at a speed dead slow.
,, B, ,, ,, 50 revs.
,, C, ,, ,, 200 revs.

For further illustrations showing the stretching of the indicator cord, see page 84.

The Influence of Drum Friction.—If the tension of the drum cord is constant throughout the stroke, the resistance offered to its motion by friction will be approximately constant, but should the tension vary considerably, it may be expected that the friction will also vary.

The general effect of friction is to impose an extra tension on the drum cord, stretching it beyond the normal elongation during the time friction is acting by an amount dependant upon the friction. The stretching of the cord produces a lag in the motion of the drum with the result that any point in the diagram is actually behind its proper position by the amount that the cord has stretched *due to friction* when the pencil is on that point.

Referring to Fig. 64 the full line represents the actual diagram taken by the indicator, and the dotted outline the true diagram that would be drawn if the drum were frictionless.

As the effect of friction is to make the actual position of a point *behind* its true position a distance equal to the

stretch of the cord, we shall have the point of cut-off B behind the real cut-off point B_1 and the point of exhaust C behind C_1. The drum will not begin to return until its spring has been allowed to recoil a distance sufficient to decrease its tension by an amount equal to the friction of

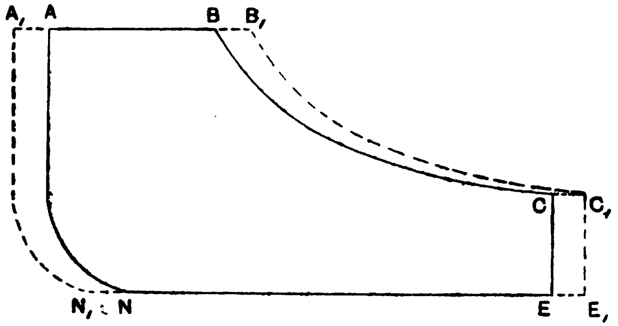


FIG. 64.—Showing Effect of Drum Friction on Diagram.

the drum, which is the amount the cord has stretched beyond the normal due to the friction of drum, and which equals E E_1 or C C_1.

During the exhaust stroke the compression point N is behind the true point N_1 and necessarily the point A behind A_1 for the same reason. The diagram as taken is shorter than that which would be produced with a frictionless drum; and Mr. Brightmore found that this shortening may amount to 7 per cent at speeds of 70 to 130 revolutions per minute with a Richards indicator, in which the drum was not well lubricated.

In his investigation, Professor Reynolds has assumed that the tension of the cord is constant, and consequently the amount of extra stretch due to the friction is constant. Let this be represented by x; then $A_1 A = B B_1 = C C_1 = x$.

Let E = number of the indicator spring.

p_a = admission pressure.

p_e = exhaust pressure.

p_m = mean effective pressure of actual full line diagram.

e = error in true mean effective pressure due to stretching of cord.

Then $p_m + e$ = true mean effective pressure in cylinder.

= mean effective pressure of dotted diagram.

Area of any diagram = mean height × length.

$$= \frac{\text{mean effective pressure}}{\text{E}} \times \text{length.}$$

or

$$\frac{\text{E} \times \text{area}}{\text{length}} = \text{mean effective pressure.}$$

Let L = length of actual full line diagram, then the mean effective pressure of dotted diagram

$$= \frac{\text{E} \times \text{area } \text{A}_1\,\text{B}_1\,\text{C}_1\,\text{E}_1\,\text{N}_1\,\text{A}_1}{\text{L} + 2x.}$$

$$= \frac{\text{E}}{\text{L} + 2x}\left(\text{A}_1\,\text{A}\,\text{N}\,\text{N}_1 + \text{A}\,\text{B}\,\text{C}\ \text{E}\,\text{N} + \text{B}\,\text{B}_1\,\text{C}_1\,\text{E}_1\,\text{E}\,\text{C}\,\text{B}\right)$$

$$= \frac{\text{E}}{\text{L} + 2x}\left\{ x\frac{p_a - p_e}{\text{E}} + \frac{p_m}{\text{E}}\text{L} + x\frac{p_a - p}{\text{E}} \right\}$$

$$= \frac{2x}{\text{L} + 2x}(p_a - p_e) + \frac{\text{L}}{\text{L} + 2x}p_m$$

Equating this to $(p_m + e)$ the other expression for the true mean effective pressure, and solving for e, we get

$$e = \frac{2x}{\text{L} + 2x}(p_a - p_e - p_m)$$

This may amount to from less than 1 per cent up to 10 or 15 per cent, depending upon the stretch of the cord,

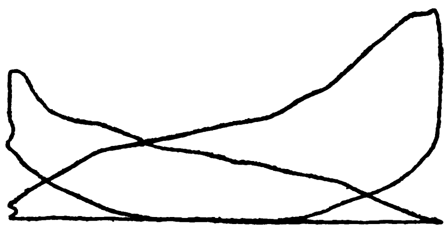


FIG. 65.—Unequal Valve Setting and Indicator Drum on Stop. Diagram $6\frac{1}{2}$ per cent short.

the pressures, and point of cut-off. Professor Reynolds found the least amount of stretch of an indicator cord to be about ·4 per cent of its length.

The actual diagram is always too small, due to the above drum friction, but pencil friction and oscillation of

the pencil mechanism have the opposite effect, and these tend to neutralise each other. With modern indicators and experienced manipulators the error in the mean effective pressure should be very small.

Indicator Cord too Long. — Care should be continually exercised to secure that the rigging of the indicator is always in good condition. A careless or inexperienced observer may permit of considerable error if this is not attended to. Fig. 65 shows at the left end of the diagrams that the cord is too long, permitting the paper drum to rest upon the stop for a certain period near the head end

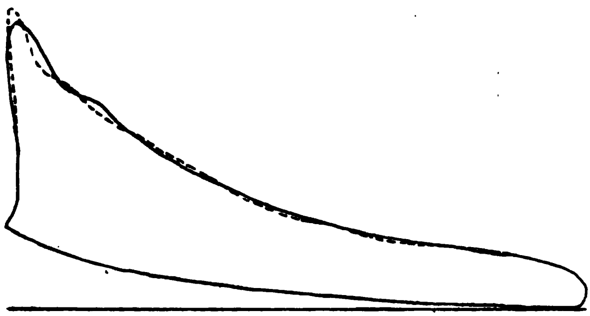


FIG. 66.—Cord too long. Diagram 5 per cent short.

of each stroke. The little indent in the admission line, and also in the exhaust toe of the other diagram, have been described by the pencil while the drum has rebounded from the stop (in the same manner that a hammer rebounds from an anvil.)

Another instance of too long a cord is shown in Fig. 66, which was taken from a 5 brake horse-power gas engine at 200 revolutions per minute, with a large Simplex indicator using a 100 spring.

In Fig. 67 this same diagram is compared with a normal diagram taken immediately afterwards, but with the correct length of cord. The dotted diagram was found to be 5 per cent short.

As the rebound from the stop occurred during ignition, when the motion of the pencil is very rapid and the spring tension small, the length of the indent extends through the whole of the explosion line.

A very instructive series of diagrams taken by an inexperienced observer (from the same engine as the

previous diagram) is shown in Figs. 68 to Fig. 74, in which the cord became longer as time went on through insecure attachment, or slipping of the cord in the cord adjuster.

FIG. 67.—Comparison of Normal with Short Diagram due to too long a cord.

On the left of the exhaust and suction stroke lines will be noticed a short vertical dotted line, which indicates the true end of the stroke. The diagram in Fig. 68 is

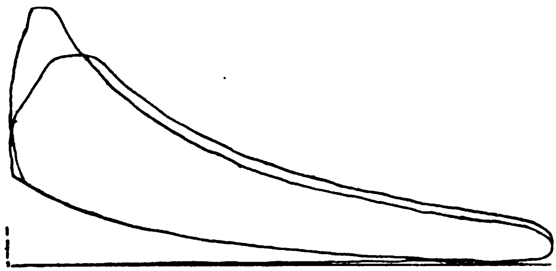


FIG. 68.—8-50 p.m.

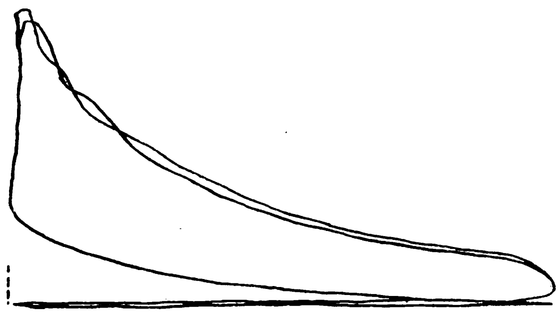


FIG. 69.—8-55 p.m. 1 per cent short.

only just the right length, while the next one is 1 per cent short. Here the concavity outwards of the ignition line leads one to suspect that the drum comes against the stop, it being entirely different from the ignition line in the previous diagram.

FIG. 70.—9 p.m. 1 per cent short.

An experienced observer would have noticed the noise of the drum on the stop unless the noise of the surrounding machinery was very great, but here it was not noticed. The diagram in Fig. 70 is similar to that in Fig. 69, but that in Fig. 71 is 2·5 per cent short, and the concavity of the ignition line is more marked.

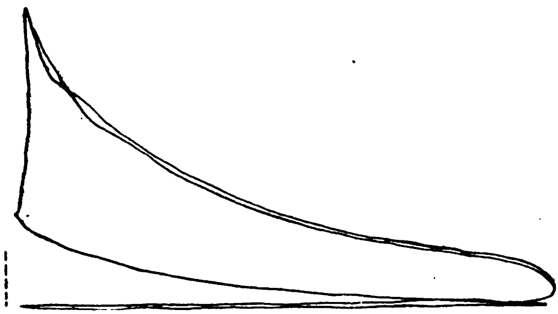


FIG. 71.—9·5 p.m. 2·5 per cent short.

The diagrams in Fig. 72 and Fig. 73 are 9 per cent short, while the knocking of the drum against the stop appears to have slightly disarranged the paper on the drum, as will be seen at the right-hand end. This is very likely to happen when the paper clips are not tight and stiff. The ignition lines in this diagram are also interesting,

as they are not coincident, and show the effect of different rates of explosion.

While Fig. 73 was being taken the noise of the drum on the stop attracted the attention of a second person, who altered the length of the cord; after which Fig. 74 was taken,

FIG. 72.*—9-10 p.m. 9 per cent short.

showing that the engine was working under normal conditions. The full load was applied by means of a rope brake throughout the run.

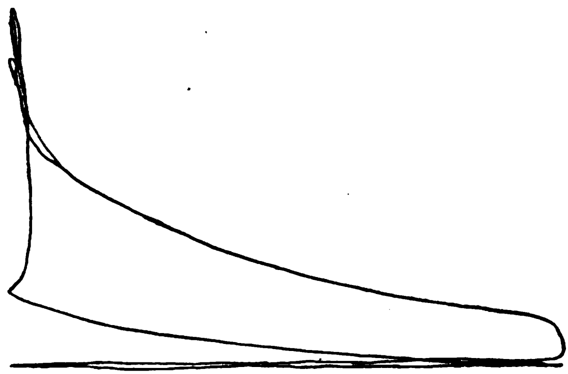


FIG. 73.*—9-15 p.m. 9 per cent short.

The diagrams in Fig. 75 were taken with the cord much too long, and purposely so, together with other diagrams of normal length taken at the same time superposed.

* These diagrams are reproduced to a slightly different scale to that of the others of the series.

Here we find the curious indent similar to that in the steam engine diagram, Fig. 68. This indent is caused by the rebound of the drum *during compression*, and the vertical cord of the indent is described by the pencil

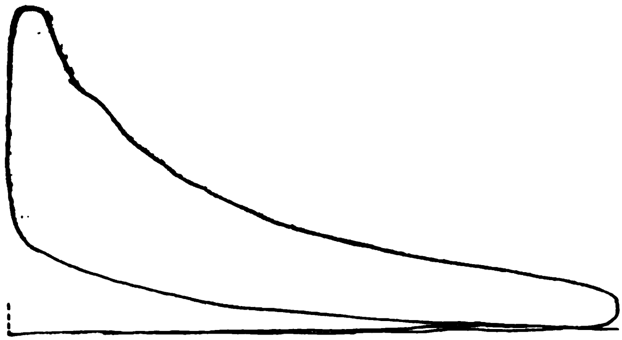


FIG. 74.—Gas Engine Diagram. 9-20. p.m. True length.

during the return of the pencil, when there is no explosion. This is shown by the arrows on the left of Fig. 76, which is a diagram taken with a 30 spring, the horizontal line at the top denotes that the piston was prevented from

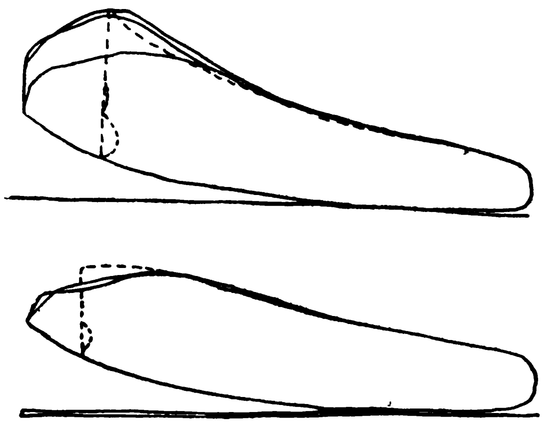


FIG. 75.—Cord too long. Gas Engine Diagram.

rising higher by the stop. This stop is generally a thin piece of brass or steel tube, cut to the right length to prevent the spring from being compressed too far. With the Simplex indicator the nature of its construction prevents

the compression of the spring exceeding a certain amount. Fig. 77 shows diagrams taken with 60 and 100 springs. The diagrams in Figs. 76 and 77 were taken within a few

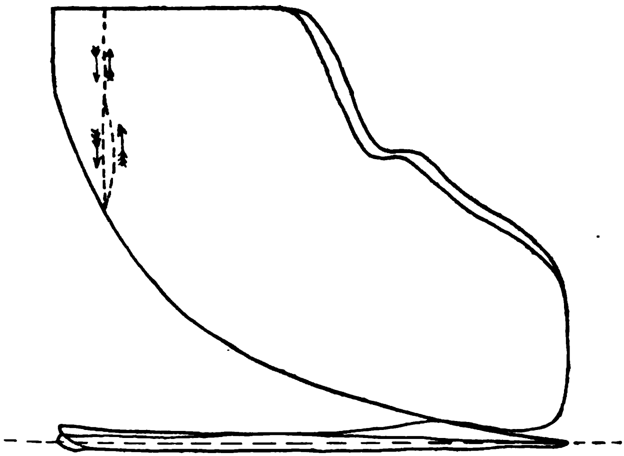


FIG. 76.

seconds of each other, which may be easily done with a Simplex indicator in which the spring is outside, and is

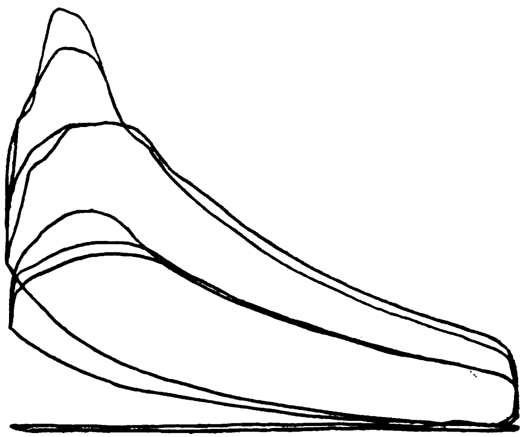


FIG. 77.

slipped into position. The load on the engine when these last diagrams were taken was about one-third of the maximum.

Displacement of Paper on Drum. — A not unusual occurrence with an indicator that has been in use some time, or with one which has not been very carefully handled, is the shifting of the paper on the drum. This may occur in the direction of the drum's axis if the paper is at all loose and the pencil presses heavily upon the paper; or it may occur along the circumference of the drum, more especially at high speeds. An instance of the former kind of dis-

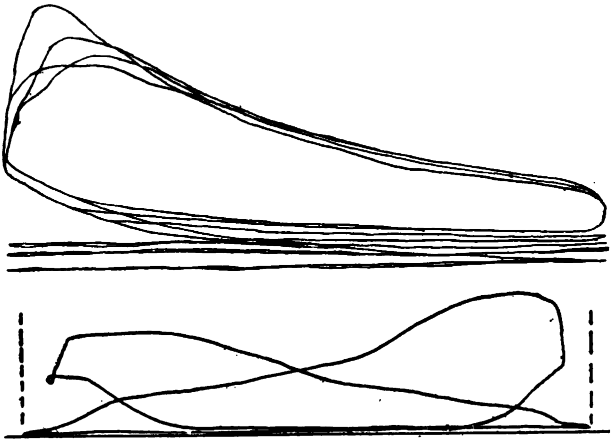


FIG. 78.—Displacement of Paper on Drum.

placement is shown in the upper part of Fig. 78, and of the latter kind in the lower part of the figure. Probably the best way of getting round this difficulty is to place a couple of elastic bands round the drum over the paper, one at each end. The clips may then be dispensed with, but great care should be exercised in arranging that the end of the paper should not come under the pencil, or the pencil mechanism may be seriously strained in tearing the paper.

FIG. 79.—Throttling Due to Indicator Cock.

Effect of Throttling in Pipe.—The diagram in Fig. 79 was taken at about 200 revolutions per minute, the dotted

line being drawn while the three-way cock was partially closed. The throttling is not excessive, but would have been considerably more at a higher speed.

Indicator Connections.—During the year 1896, Prof. Goss, of Lafayette, Indiana, carried out some experiments on the effect of long indicator connections on the diagram; and in the same year presented the results of his work to the American Society of Mechanical Engineers, from whose proceedings the figures immediately following have

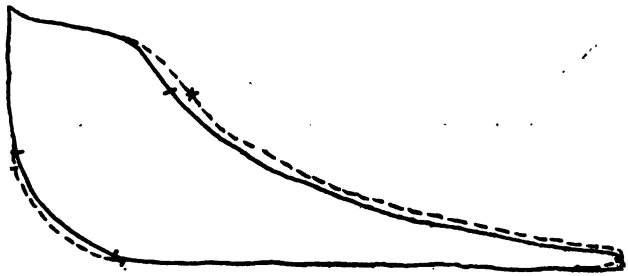


FIG. 80.—5 ft. Pipe ; 200 Revolutions ; ¼ Cut-off.

been extracted. The method of the experiment was to take eight diagrams with an indicator attached direct to the cylinder, and another eight diagrams with a similar indicator connected to the cylinder by different lengths of pipe. The average of each set was used for comparison. The full line diagrams, Figs. 80 to 84, are those taken with the indicator fixed in the cylinder, and the dotted diagrams those taken with the long pipe.

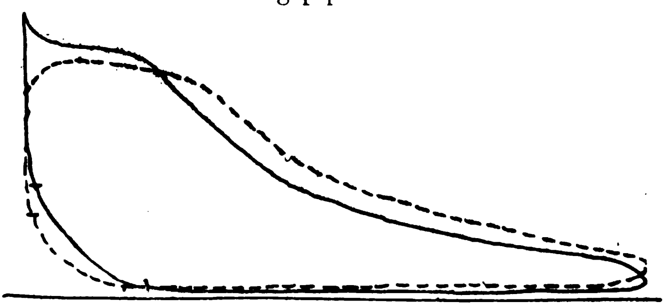


FIG. 81.—10 ft. Pipe ; 200 Revolutions ; ¼ Cut-off.

It was found in Figs. 80 to 85 that the longer the pipe the later did each event appear to occur after it ought to have occurred. This is especially noticeable in the

exhaust line of Fig. 80. The marked points in both sets of diagrams indicate the amount of lag.

The main conclusions drawn from the experiments were that:—

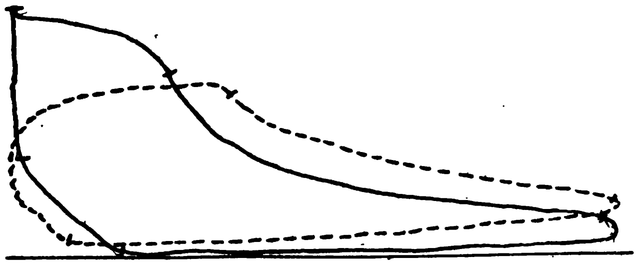


FIG. 82.—15 ft. Pipe; 200 Revolutions; $\frac{1}{4}$ Cut-off.

(1) An indicator should be attached direct to the cylinder where possible.

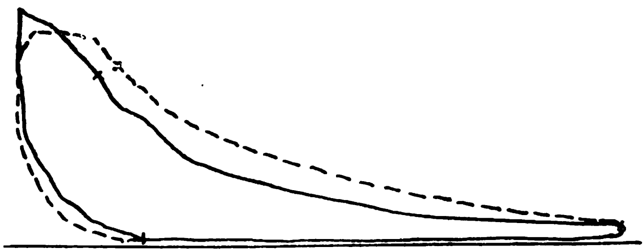


FIG. 83.—10 ft. Pipe; 200 Revolutions; $\frac{1}{8}$ Cut-off.

(2) Any pipe connection is likely to effect the diagram, a very short pipe may produce a measurable error, but a

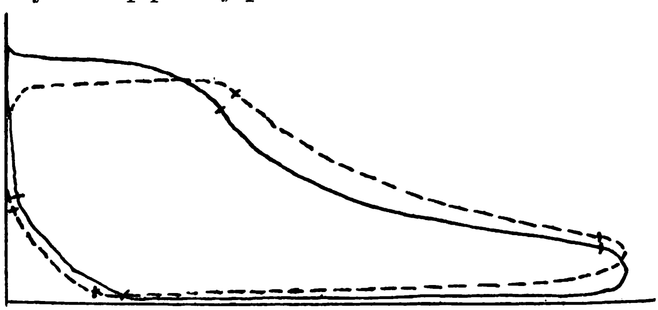


FIG. 84.—10 ft. Pipe; 200 Revolutions; $\frac{1}{3}$ Cut-off.

length of 3 ft. or over may be sufficient to render the card valueless except for rough or approximate work.

(3) The effect of a pipe connection is to retard the pencil action, being more pronounced with an increase of speed or expansion.

(4) Within limits the indicated power is increased by pipe connections, and conclusions, as to heat exchanges, are unreliable when based upon diagrams so taken, even though the pipe is short.

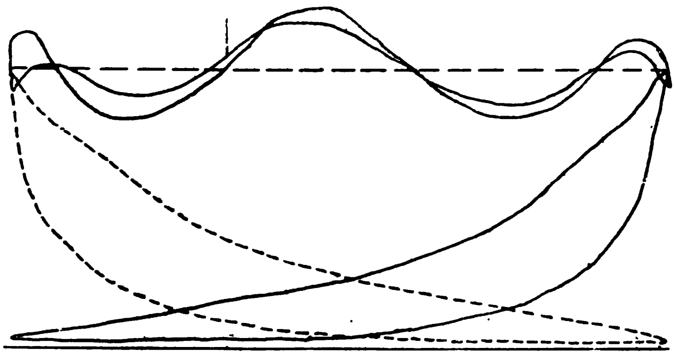


FIG. 85.—Excessive Oscillation in Valve Chest Diagram.

The diagrams, Fig. 85, were taken with a Tabor indicator from a locomotive, the connection to the valve chest being long. The boiler and steam chest mean pressure was 156 lb.

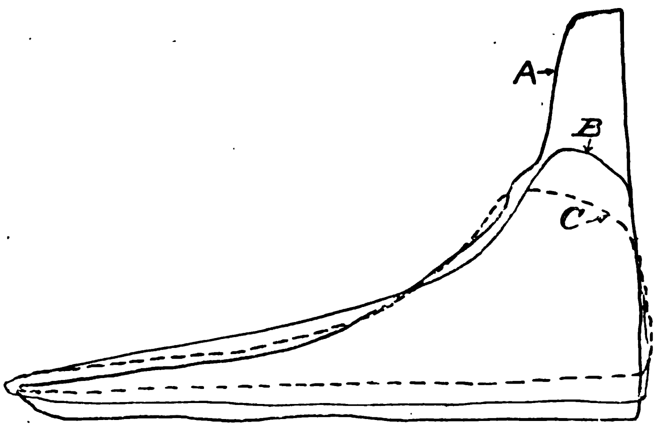


FIG. 86.—Showing Effect of Long Indicator Pipes.

per square inch, cut off 17 per cent of the stroke, and revolutions 238. The oscillations could not have been due to the indicator, or they would have appeared in the cylinder

diagrams. Other diagrams taken at different speeds show some oscillation, but not so much, as in this one. Probably, the speed was such that the natural period of oscillation of the column of wet steam in the pipe connection approached that of the taking of steam by the cylinder from the steam chest.

The influence of long pipe connections is well shown in Fig. 86. The diagram A was taken with an indicator attached direct to the cylinder, while B was taken with an indicator situated midway between the ends of the cylinder and connected by a three-way cock. The dotted diagram was taken some time afterwards, when the pipe was probably partially choked. It was taken from a Corliss engine at a speed of 44 revolutions per minute.

Stretching of the Indicator Cord.—Mr. Brightmore experimented upon the stretching of the indicator cord, and used for the purpose an apparatus in which a make and break of electric contact occurred at intervals, which divided the stroke into ten equal parts. A contact piece was attached to the crosshead, which passed over a series of metallic pegs secured in a wood strip and connected through a battery to a Rumkorf induction coil. The high tension terminals were connected to the pencil frame and to the drum frame. As the contact piece on the crosshead passed over a peg, the low tension current from the battery induced a high tension current through those terminals attached to the indicator, with the result that the current

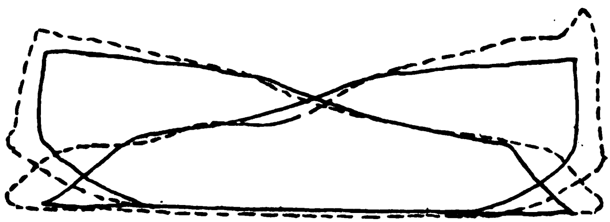


FIG. 87.—Stretching of Cord at High Speed.

pierced the paper previously placed on the drum and recorded how much the drum lagged behind the piston. With fine wire instead of cord this lag was very small, but with cord it was considerable, amounting to 5 per cent (at 130 revolutions per minute) in some parts of the diagram. Of course this lag was permitted by the stretching of the cord; with wire the lag was almost imperceptible. The

distortion of the indicator diagram through the stretching of the cord can be easily shown by taking two diagrams upon the same card; one at a slow speed and another at a high speed; and, as previously pointed out, the magnitude of the stretching will depend upon the length of cord, the initial tension of the cord, the speed of reciprocation and the moment of inertia of the drum.

Fig. 87 shows two sets of diagrams taken in the above manner. The full line diagrams were taken at about fifty revolutions per minute, and the dotted diagrams a few seconds after under similar conditions except the speed, which was increased to 350 revolutions per minute. The drum spring was given a half a turn from being dead slack

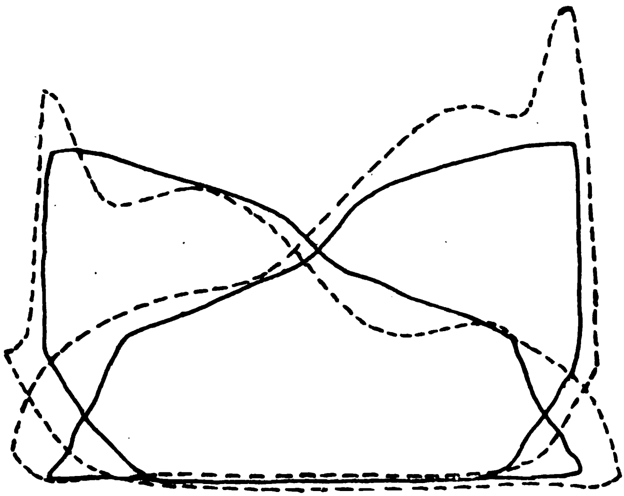


FIG. 88.—Stretching of Indicator Cord.

before either diagram was taken, and a 60 spring was used. The dotted diagrams are 13 per cent longer than those taken at slow speed.

The drum spring was afterwards given another quarter of a turn and the diagrams in Fig. 86 were then taken under the same conditions as those of the last figure, except a 30 spring was used. The dotted diagrams are here 11 per cent too long. Probably the paper shifted vertically very slightly before taking the left-hand diagram. This may have been due to the jar of the drum as it came against the stop at the higher speed. The shape of the left end of the head end dotted diagram should be noted,

it resembles, to a certain extent, that obtained at a different speed with a gas engine when the indicator drum was on the stop, as it was in this case due to the stretching of the cord.

After these diagrams were taken the drum spring was put back to one quarter of a turn from dead slack, and the diagrams of Fig. 89 were taken. In this case the drum was always some distance off the stop. The full-line diagrams were taken at a very slow speed, after which the

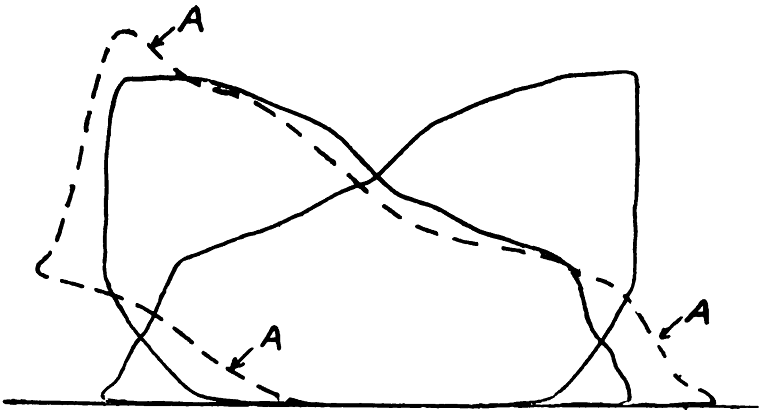


FIG. 89.—Diagram Showing Stretching of Cord.

speed was increased and the dotted diagram taken at about 300 revolutions per minute. As this speed was increasing the cord hook disengaged itself before another diagram could be taken. The length of cord was about 3 ft., and the amount of stretch at the higher speed was

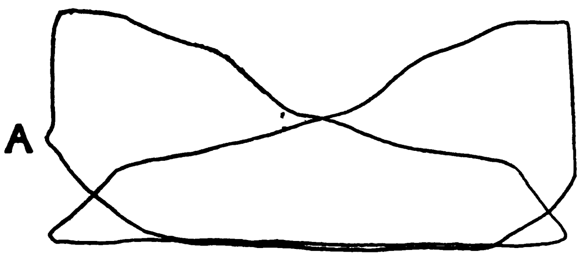


FIG. 90.

24 per cent of the length of the diagram. The indicator used was a large slow-speed "Simplex." The left hand end of the diagram in Fig. 90 contains a protuberance at A

something like that at the same end of the dotted diagram, Fig. 89, and due to the same cause, namely, the stretching of the cord, due to the inertia of the drum. The diagram was taken with a high-speed instrument at 350 revolutions per minute, but with the drum spring not sufficiently wound up. The protuberance is at the head end of the diagram, and at the instant the tension of the cord was a minimum.

Copying Diagrams.—When copies of diagrams are required, too much care cannot be taken in tracing over the lines, as it is very easy to follow the wrong one. This is the more likely if the diagrams are not quite new, as

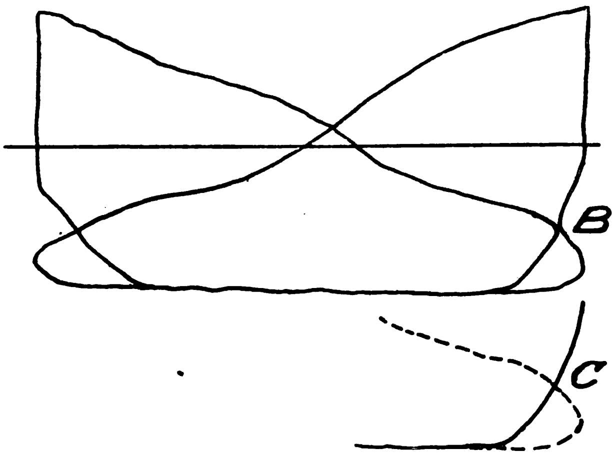


FIG. 91.

they often fade. Fig. 91 gives an instance of incorrect tracing at B, the correct indication being shown at C below.

CHAPTER VI.

PRELIMINARY ANALYSIS OF THE DIAGRAM.

The Cardinal Points of a Steam Engine Diagram.—A typical pair of good diagrams taken from a Corliss engine is given in Fig. 92, while diagrams taken from an engine fitted with an ordinary piston slide valve are shown in Fig. 93, similar letters in both figures referring to the same points.

Steam begins to enter the cylinder at the end of compression at Q when the engine piston is at the end of its stroke, compelling the pencil to trace the vertical line above Q.

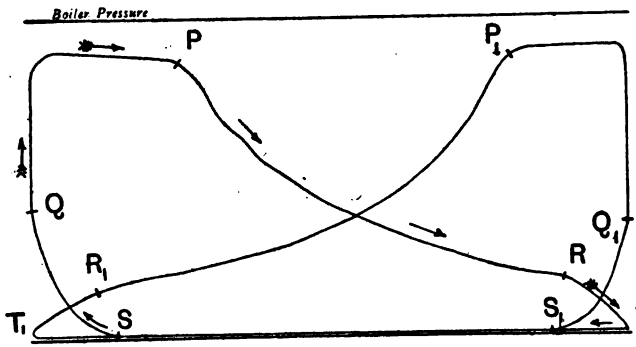


FIG. 92.—Typical Corliss Engine Diagram (non-condensing), 60 revolutions per minute.

As the engine piston moves down the cylinder, the pencil describes the nearly horizontal line up to P (called the steam or admission line), at which point—the point of *cut-off** the supply from the boiler is cut off by the closing of the steam port by the valve. The steam that is in the cylinder now expands in volume, and consequently decreases in pressure, the pencil tracing out the line P R (called the *expansion* curve). At R the exhaust valve begins to open and allows the steam to pass out into the

* The actual point of cut-off has been shown to be at the *point of inflexion*, a little distance beyond the first bend in the steam line.

atmosphere, but as the port opens comparatively slowly, the steam pressure is *gradually* reduced, the pencil describing the line R T.

From T to S the exhaust port remains open, and the returning engine piston pushes the cylinderful of

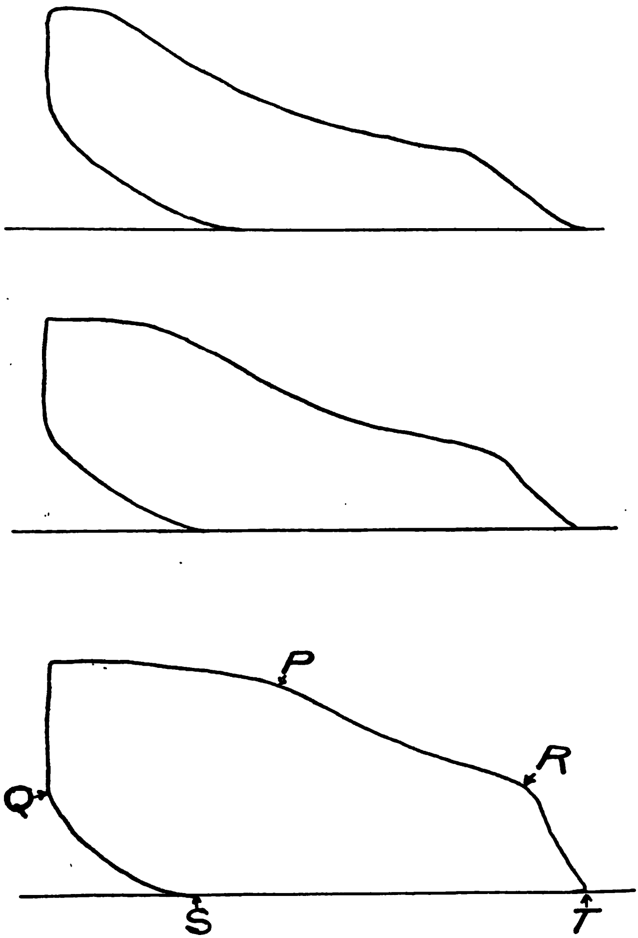


FIG. 93.—Typical Diagrams from a Non-condensing Engine with Ordinary D-slide Valve.

steam (that remains after the first puff) into the atmosphere. At S the exhaust port closes, and the steam that

is left in the cylinder is compressed till the piston gets to the end of its stroke, when the steam port opens at Q again, and the cycle of events is repeated.

The diagrams in Fig. 92 were taken when the engine was running at 60 revolutions per minute; and at this slow speed, with ports properly designed, the steam line is nearly horizontal. The steam has sufficient time to enter the cylinder, and maintain the pressure approximately equal to that in the steam pipe or separator near the stop valve.

At the higher speed of 200 revolutions per minute (Fig. 93) the steam line slopes towards P more than in the previous figure because the steam cannot enter the cylinder in sufficient quantity during the time the valve is open; and, further, when the pencil is nearing P the steam port is closing more and more, giving less and less area through which the steam can pass, while the piston is increasing its velocity.

The two upper diagrams in Fig. 93 were taken from the same engine when cutting off steam at one-quarter and three-eighths of the stroke respectively, the lowest diagram being taken when cutting off at half stroke. The variations in the cut-off were effected by shifting the eccentric across the shaft* keeping the lead constant. In the uppermost diagram the cut-off appears earlier than one quarter of the stroke. This is caused by the opening of the port being very small at this cut-off on account of the reduced travel of the valve, and consequent throttling or wire-drawing of the steam.

The amount of wiredrawing, or reduction of pressure, is due to speed as well as to restricted openings, for the greater the piston speed the more steam is required in a given time to follow it up. It is found more or less in all diagrams, as will be seen in many of the following examples. In Fig. 92 the steam line up to P is considerably below the boiler pressure line. This is due to the length of the steam pipe between the engine and the boiler, it being, in this case, 117 ft., and contained ten right angle bends. Had the pipe been only a few feet long, and of ample sectional area, the steam line would have been approximately coincident with the boiler pressure line.† The exhaust line in Fig. 92 is above the atmospheric line, showing a little back pressure. The exhaust port opening was probably small, or the exhaust pipe rather long.

* See Fig. 195. † See Figs. 147 to 149.

Hints on Taking Indicator Diagrams.—See that the different parts of the indicator are in good working order, and that the drum spindle and piston are properly lubricated. Cylinder oil may be used for the latter, or special gas engine oil. Do not use inferior oil. Many experimenters oil the piston again after taking from six to ten diagrams.

See that the indicator is properly rigged up together with its reducing motion, and previous to running the engine, obtain the correct length of cord. It is not advisable to do this while the engine is at work unless the operator has had a fair amount of experience, or the indicator may be unintentionally damaged.

Put the paper on the drum carefully and do not strain the clips. Should these not hold the paper tightly, they should be taken off and bent inwards. Do not take the paper drum off its spindle any more often than is really

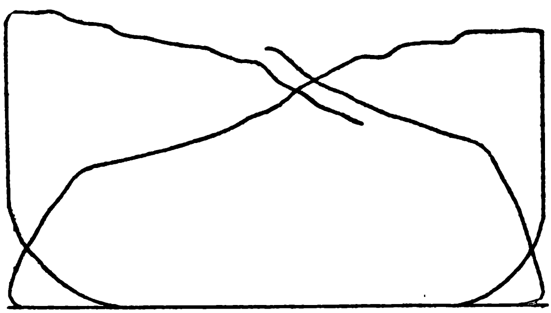


FIG. 94.—Showing Necessity for Holding Pencil on paper for more than One Complete Revolution.

necessary, as it tends to get loose. Adjust the tension of the drum spring so that the drum shall not over run at the end of its stroke more than is absolutely necessary.

Keep the pencil fairly sharp and do not allow it to press heavily on the paper. A heavy line may cause the paper to shift and distort the diagram; besides, the friction caused thereby produces an incorrect diagram.

Before attempting to take a diagram, blow steam through the drain hole in the indicator cock until no water, but only blue steam, is visible. The indicator is now warmed up and is ready to be used to take a diagram, but turn the cock round so that there is no steam going to

waste before bringing the pencil to the paper. Take the atmospheric line while the indicator is hot.

A leaky piston is better than one that is too tight a fit.

If the diagrams are taken for the purpose of finding the indicated horse power, the pencil should be kept on the paper for from four to ten or more revolutions. Failure to do this may cause the diagram shown in Fig. 94, which was taken with a constant load and is produced by the hunting of the governor, which in this case acted upon a throttle valve.

If the same indicator is used for both ends of the cylinder, the connection being made through a three-way cock, note which diagram belongs to the head and crank ends respectively, and label them on the paper. The atmospheric lines should be drawn by hand before the diagrams are taken by pulling the indicator cord.

A false atmospheric line may be drawn, as shown at A A, Fig. 95, when using certain makes of indicator cocks, if the operator is not careful to turn the plug of the cock, so that

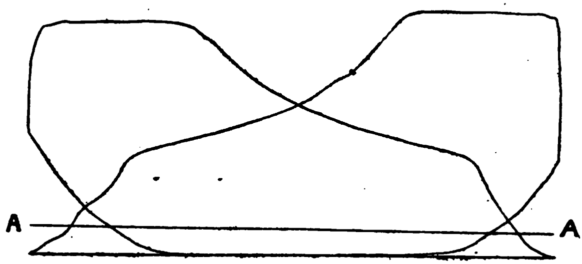


FIG. 95.—Non-condensing Engine Diagram, False Atmospheric Line.

the cylinder is *entirely shut off by the cock*, and at the same time the indicator is in communication with the atmosphere through the little drain hole in the body of the cock. The false line is caused by the cock imprisoning a certain amount of steam in the indicator cylinder.

After the paper has been removed from the drum, the date, time, and number of spring should be noted upon it, together with the number denoting the order in which it was taken, and the initials of the person who actually took the diagram. If it is only desirable to take a couple of diagrams to see how the engine is working, then the following additional information should be written on the back of the card :—

Kind of engine..
Makers ....
Which cylinder ....
Diameter of cylinder.......Diameter of piston rod.......
Length of stroke...............Boiler pressure...............
Speed....................Barometer ....
Vacuum by gauge.......................inches of mercury

and any other information which may be thought necessary.

In taking the indicator down, be careful not to bend the pencil gear. Clean it thoroughly and oil it before putting it in its box. See that the indicator is properly placed in its box and that the lid of the box does not press hard upon any part of the indicator when closing the lid, otherwise some part of the instrument may be bent.

It is advisable to use a wire in the place of the cord if it is unavoidably long.

Should the stroke of the engine be very long, or if the ordinary reducing gear cannot be easily arranged, the following gear may sometimes be used with advantage. It is one of the perfect type.

A Couple of Useful Reducing Gears.—A circular disc W, Fig. 96, is fixed eccentrically upon the crank shaft, which is shown shaded with its centre in the centre line of the crank, and on the same side of the shaft. The throw of this disc is equal to half the required length of the diagram to be taken. Butting against the periphery of the disc is a roller V situated in one end of the sliding rod K, which is supported in the two brackets S and R. A spiral spring pressing against a collar on the rod K maintains the roller in contact with the disc, and thus the roller and rod K partake of a reciprocating motion. The end L of the rod K is much of the shape of the claws of a claw-hammer, between which may be slipped the cord Q of the indicator; a small piece of wood or insulating fibre secured by a knot preventing the cord from slipping between the claws at M. A plan and end elevation of these are shown at M_1. It will be found convenient to slip the end of the cord over the hook F, so that it is quite handy when it is required to place the cord between the claws M.

A little pulley sheave may be placed on the end of the sliding rod instead of the claw end, and the end of the cord may then be slipped over a hook fixed to some stationary part, as shown at N. The throw of the eccentric

disc in this case requires to be only one-fourth of the length of the diagram. This gear gives perfect reduction when

$$\frac{\text{sum of radii of disc and roller}}{\text{length of connecting rod}} = \frac{\text{throw of disc}}{\text{length of crank}}$$

$$= \frac{\text{length of diagram}}{\text{length of stroke of engine.}}$$

Referring to the upper part of Fig. 96.

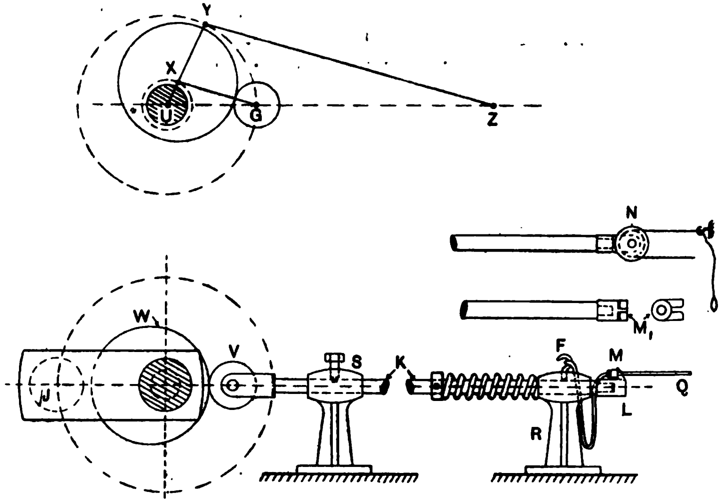


FIG. 96.—A Reducing Gear used by Messrs. Hick, Hargreaves, and Co.

Y is the crank pin centre
X ,, disc centre
G ,, roller centre
U ,, centre of crank shaft.

X G = radius of disc + radius of roller, and U X G is a reproduction of U Y Z to a reduced scale, when X G is parallel to Y Z. This will always be the case when—

$$\frac{XG}{YZ} = \frac{UX}{UY} = \frac{\text{throw of disc}}{\text{length of crank.}}$$

But X G = radius of disc + radius of roller,
and Y Z = connecting rod length,

hence the relation given above.

The author is indebted to Mr. J. G. Hudson, M.I.C.E, for the above illustration.

A handy reducing gear, made of aluminium, by the Lippincott Company, of Benedict Building, New York,

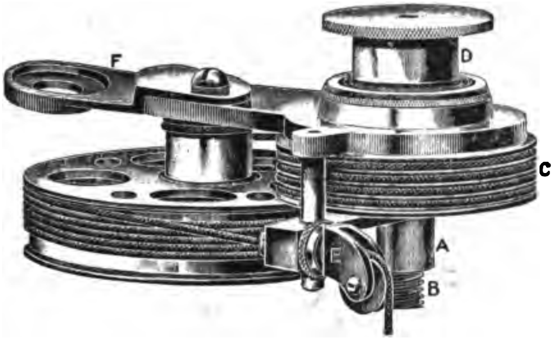


FIG. 97.—Aluminium Reducing Wheel.

is shown in Fig. 97. It is made to be attached to the drum spindle of existing indicators, or to be

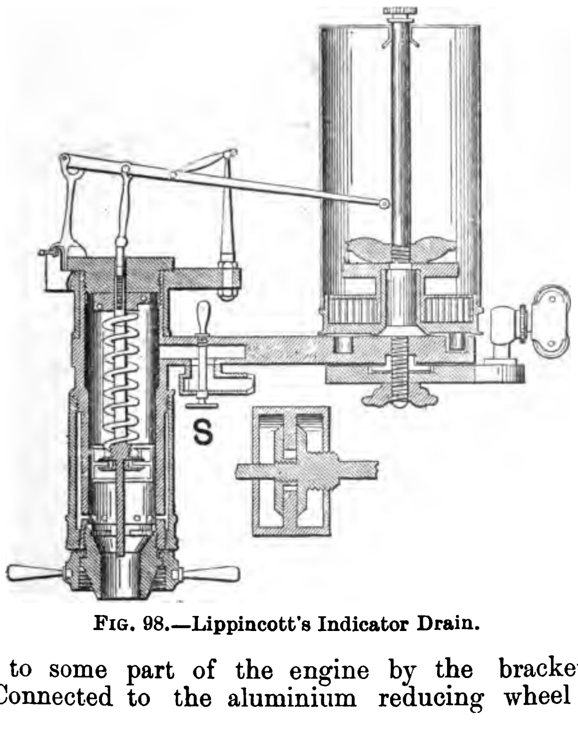


FIG. 98.—Lippincott's Indicator Drain.

fixed to some part of the engine by the bracket-arm F. Connected to the aluminium reducing wheel is a

small pulley, round which is wound a braided cord which runs to the spring-wheel C. This wheel turns in conjunction with the small pulley D, round which is wound the cord running to the paper drum. There are a number of different pulleys supplied with the instrument, so that any usual reduction may be accomplished. The chief feature in the attachment is the impossibility of the cord on the main wheel riding upon itself, although it may make three or four consecutive turns. The guide pulley and bracket E are made to slide up the rod by the rotation of the screw B in the bracket A, and thus the cord is always properly coiled upon the reducing wheel.

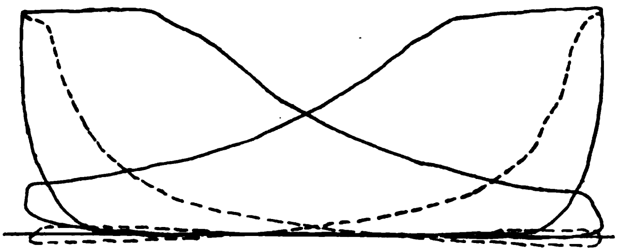


FIG. 99.—To Show Effect of Cut-off Valve with Separate Distribution Valve.

Another improvement by the same makers is the addition of a swivelling nozzle to the drain holes at S, Fig. 98, so that any leakage may be effectually guided away from the operator. It is adjustable.

CHAPTER VII.

DIAGRAMS SHOWING VARIATION OF LOAD.

Variation of Load.—Stationary engines have their speed regulated by the action of the governor on the slide valve, making it cut off at different points of the stroke; or by the action of the governor on a throttle valve by which the initial pressure of steam is varied to suit the work being done.

These two methods of governing are often called "expansion governing" and "throttle governing" respectively.

Fig. 99 shows the action of the separate cut off or expansion valve, dotted lines being obtained at light load.

Diagrams from a Corliss engine showing variations in load are given in Figs. 100 and 101.

In the next figure diagrams are given which were taken from a single cylinder Corliss engine while working

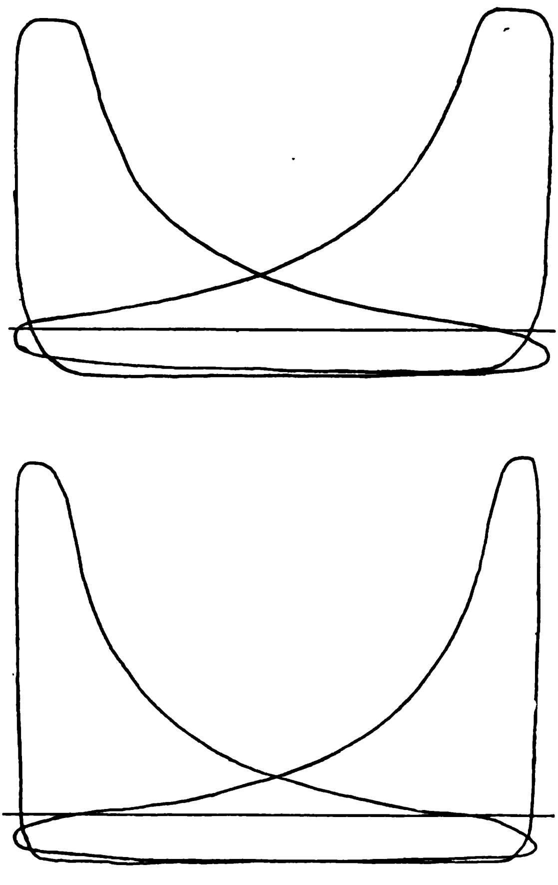


FIG. 100.—Diagrams from a Simple Condensing Engine with Corliss Valves, Showing Variation of Load.

with a load much above the normal. The cylinder was 16 in. diameter and the stroke 3 ft., the speed being 75 revolutions per minute and the boiler pressure 80 lb. per square inch.

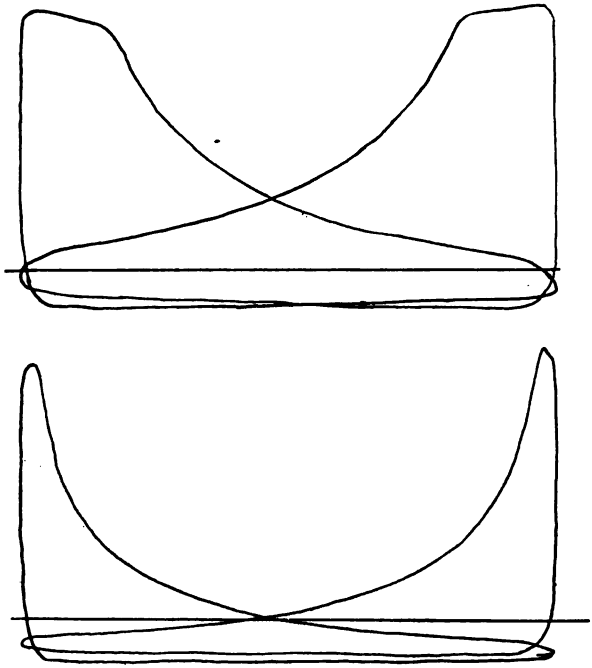


FIG. 101.—Diagram from an Engine with Corliss Valves, Showing Variation of Load.

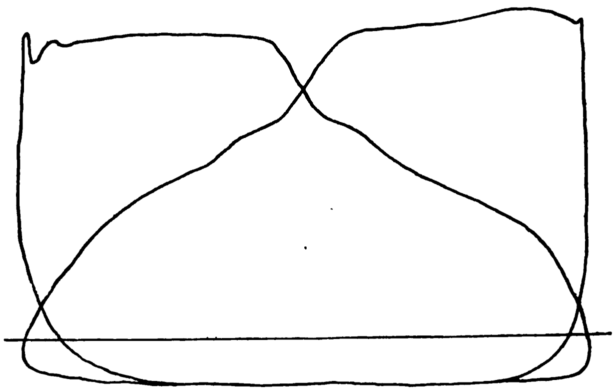


FIG. 102 —Case of Heavy Overloading. Normal Load, 90 Indicated Horse Power; Actual, 174 Indicated Horse Power.

Another case of overloading is given in Fig. 103. The full economical load for which the engine was designed was 385 indicated horse power, but when the diagrams were taken it was developing 601 indicated horse power during ordinary work without any alteration to the valves.

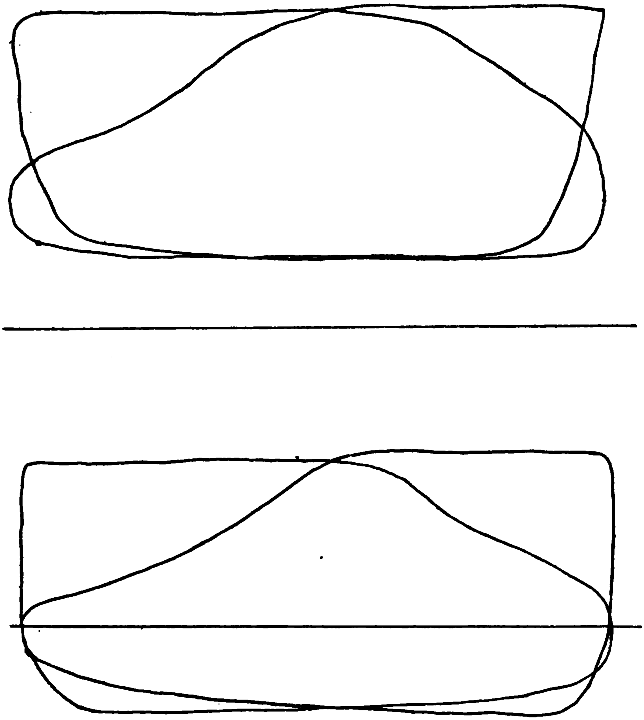


FIG. 103.—Compound Corliss Engine, Heavily Overloaded.

The boiler pressure was 120 lb. per square inch, the cylinders being 16 in. and 30 in. by 3 ft. stroke, and the speed 100 revolutions per minute.

The diagrams in Fig. 104 were taken from an electric railway power station engine with cylinders 11 in. and 20 in., by 15 in. stroke, while running light. The middle diagram was taken with a number 10 spring from the receiver; the other springs being high pressure, 60; low pressure, 20.

Excessive compression will be noticed in both cylinders, causing loops at the top of all the diagrams. These loops

represent negative work, that is, work done by the pistons on the steam. Loops of a similar nature will be found at

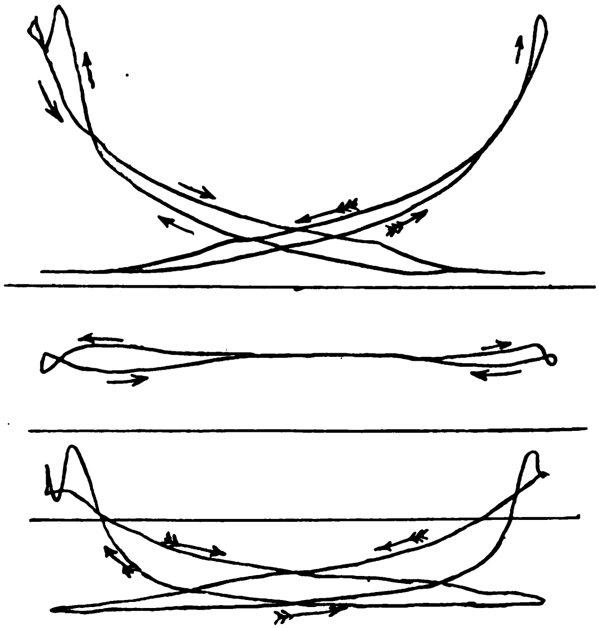


FIG. 104.—Compound Tandem Electric Railway Engine.

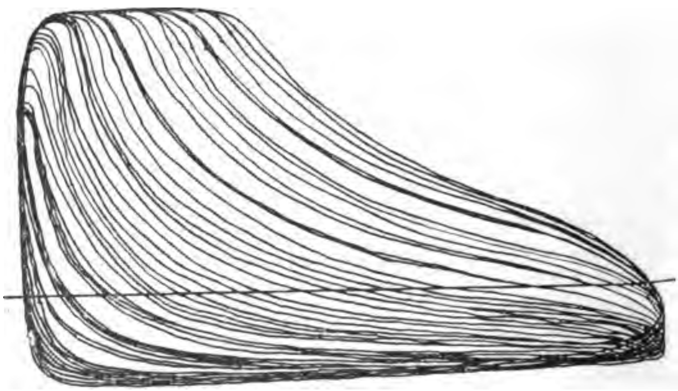


FIG. 105.—From a Single Cylinder Corliss Engine Driving a Rolling Mill. Roughing and Finishing Passes.

the other end of the dotted diagrams of Fig. 99. Here the steam is expanded below the exhaust pressure, during

which the crank is dragging the piston after it, and a certain amount of work is done by the crank upon the atmosphere. The compression shown in Fig. 104 is excessive, and would be much reduced with some load on.

The diagrams in Fig. 105 are interesting as showing the large variation of load which occurs in a rolling mill. The diagrams shown are not nearly all of those described by the

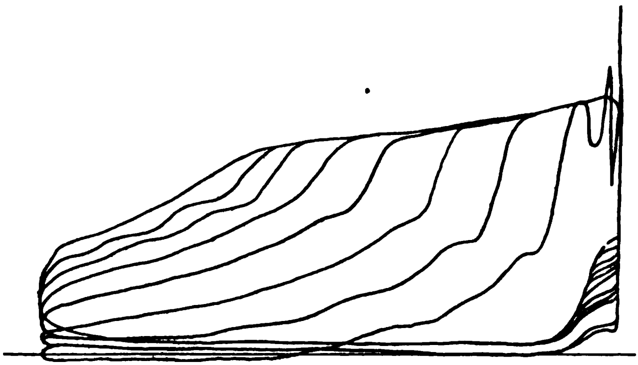


FIG. 106.—From a Millwright's Engine Fitted with Messrs. Hick, Hargreaves, and Co's. High-speed Corliss Gear.

pencil during one set of roughing and finishing passes. Fig. 106 gives a set of diagrams taken from one of Messrs. Hick, Hargreaves, and Company's engines fitted with their new high-speed Corliss gear. The governor was moved slowly over the whole range by hand, while the diagrams were taken, the object being to show the extensive range of cut off with this gear. This very wide range is shown better in Fig. 107, which contains diagrams taken from the high-pressure cylinder of one of the engines supplied to the Leeds Electric Lighting Works by the same firm. As the valve gear which produced these diagrams is interesting, a couple of illustrations of it are given below. A perspective view of a pair of engines fitted with this gear, is given in Fig. 108, and a detail of the valve gear is shown in Fig. 109. Reverting to Fig. 108, it will be seen that the governor G partially rotates the spindle L during a change of speed, and this rotation moves the rod K longitudinally, altering the position of cut off. The four valves are operated (with the exception of cutting off steam) by the single wrist plate shown in the figure,

worked by a single eccentric. The cut-off cams are operated by a small separate eccentric, which drives the rocking lever H and the rod J.

Passing on to Fig. 109, the rods P and Q are the rods J and K of the previous figure. The wrist plate operates the rocking lever B by means of the link A. The rocking lever is connected to the sleeve M by the link C, which also carries the pin G,

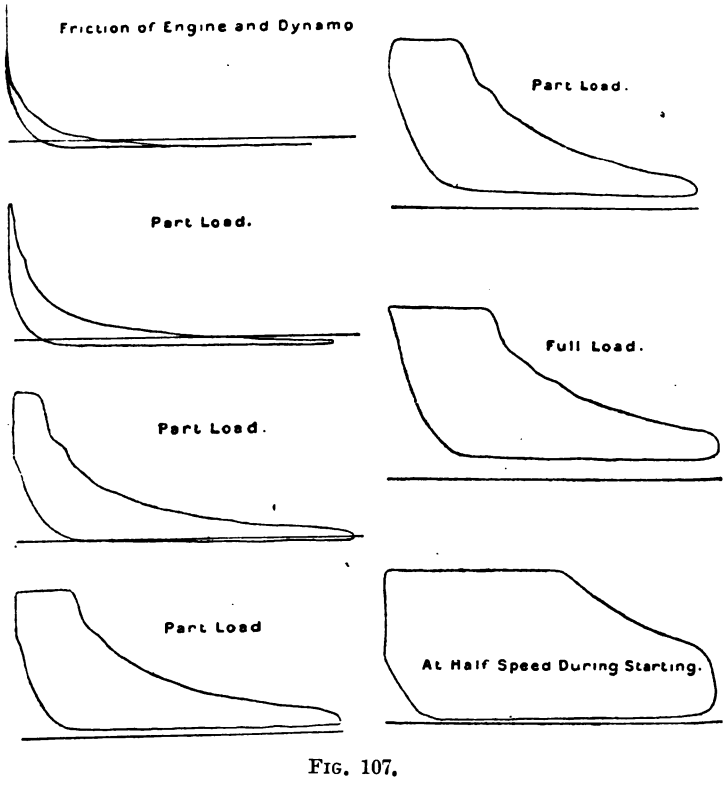


FIG. 107.

upon which the cut-off detent T is supported. Inside the sleeve M slides the valve rod which is attached to the dashpot piston at one end and the valve spindle arm at the other. A spiral spring C S forces the dashpot piston home as soon as the detent T disengages with the valve rod. It will be noticed that the spring is compressed during the inward stroke of the sleeve M, and remains

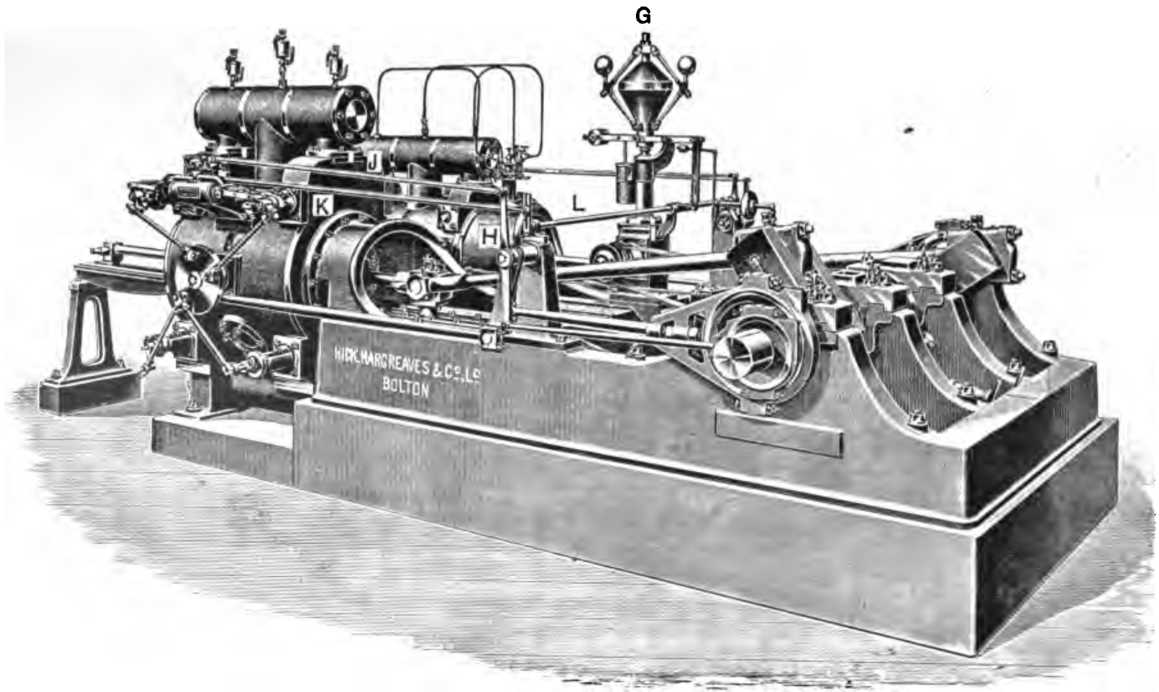


FIG. 108.—ENGINE FITTED WITH MESSRS. HICK, HARGREAVES, AND CO'S. SPECIAL HIGH-SPEED CORLISS VALVE GEAR.

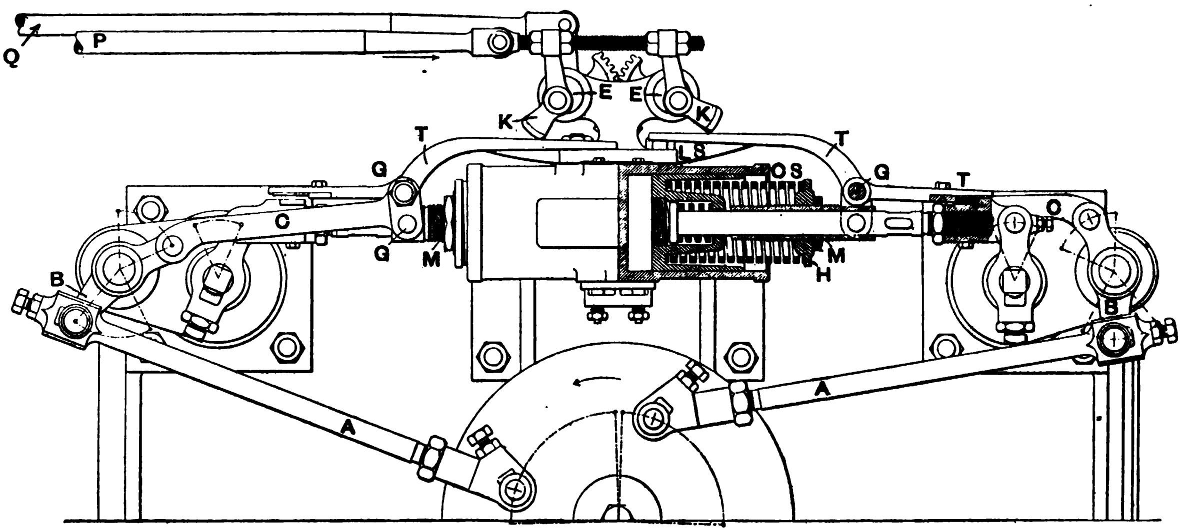


FIG. 109.—DETAIL OF HIGH-SPEED CORLISS GEAR, MESSRS. HICK, HARGREAVES, AND CO.

compressed until the detent is released sometime in the next stroke so that the driving force used to close the valve is always the same, no matter what may be the cut-off.

Two little cams K, operated by the rod P, which is made to reciprocate by the small cut-off eccentric, alternately press upon the inner arms of the detents T, releasing the valve rods and allowing the spiral spring to return the valve rods to their zero position, thereby cutting off steam. The rod Q (coming from the governor) is connected to the two eccentrics E E just above the dashpot. These eccentrics when rotated either raise or lower the cams K, thus bringing them into contact with the

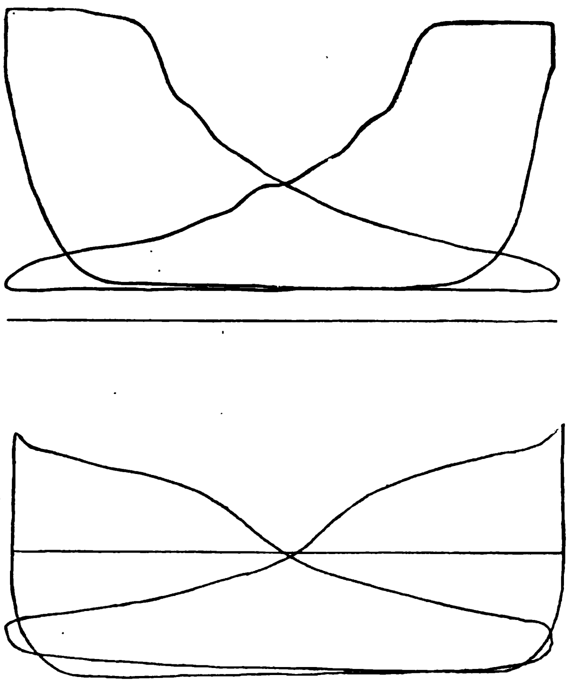


FIG. 110.—Condensing.

detents T later or earlier respectively, and consequently cutting off later or earlier as the case may be. The effectiveness of the gear may be judged from the diagrams already given. It may be interesting to know that similar engines

at the Leicester Electricity Works, having cylinders 16 in. and 30 in. by 36 in. stroke, are run at 96 revolutions per minute, and others with cylinders of 11 in. and 20 in. by 24 in. stroke are run at 118 revolutions per minute.

The engines at Leeds are run with a jet condenser, but in case of emergency can be made to exhaust into the atmosphere and still develop full power by only shutting off the condenser connection, and opening that to the atmosphere. Fig. 110 gives a set of diagrams when working condensing at 386 indicated horse power, and Fig. 111

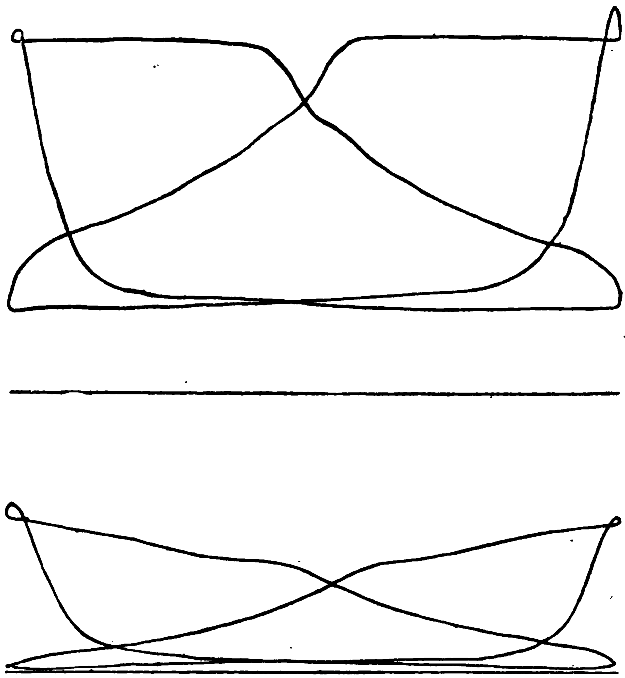


FIG. 111.—Non-condensing.

shows another set after the change over had taken place, the indicated horse power being 406. The respective steam consumptions were 14·63 lb. and 20·31 lb. per indicated horse power per hour.

Fig. 112 shows a set of diagrams taken from a compound condensing engine working under normal load, while the next figure shows diagrams taken from the same

engine when running non-condensing without altering the valves. Arrows have been added to the high-pressure diagrams to make them more intelligible. A negative horse power was developed in this cylinder when working non-condensing.

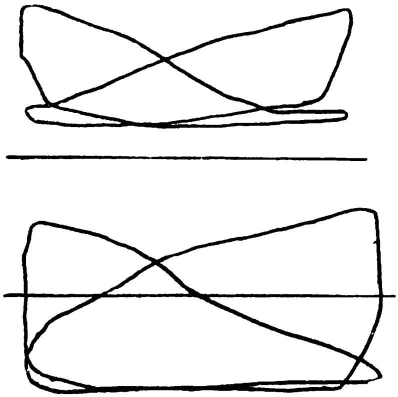


FIG. 112.—Diagrams from a pair of Compound Engines Condensing.

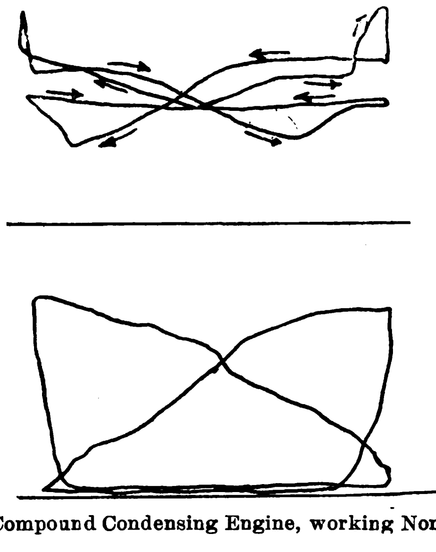


FIG. 113.—Compound Condensing Engine, working Non-condensing.

In the above instances of load variation the steam supply has been regulated automatically according to the load. In the few sets of diagrams given below the steam

supply has been fixed by the ratio of expansion, the cut-off being determined by the position of the reversing

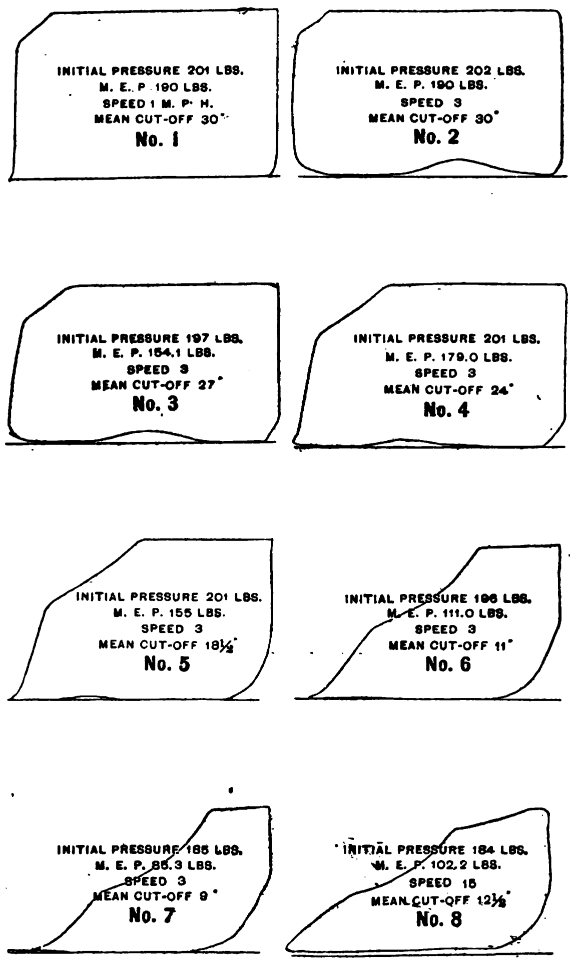


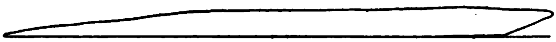
FIG. 114.

lever in the quadrant rack, the gear used being the Stephenson link motion, fitted to a locomotive.

The diagrams in Fig. 114 were taken from an American locomotive of the Mastodon type, with piston valves fitted with packing rings. The information printed upon each diagram shows how well the piston valves distributed the steam to the cylinders, and the area of each diagram

M. E. P. 12.5 LBS.

M. E. P. 13.7 LBS.



Cards from Engine Running Light.
M. E. P. = 6.5 per cent. of Initial Pressure
FIG. 115.

gives an idea of the load at the time it was taken. It must be borne in mind that the horse power does not necessarily indicate the load on the engine. Horse power is made up of two factors—effort and speed. It is the effort which is a measure of the load which is independent of speed. In locomotive work a large horse power is never associated with maximum effort.

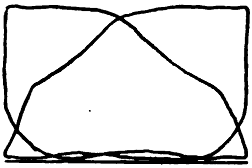
While these diagrams are before us it would be as well to notice one or two conspicuous features in them.

The rise or hump in the exhaust line is large to begin with, but decreases as the cut-off is made earlier in the stroke. This is due to the exhaust from the companion cylinder, causing back pressure. This is greater as the pressure at the beginning of the exhaust is greater, but when the exhaust pressure is decreased by an earlier cut-off, or the speed is increased, causing a more gradual exhaust, the hump is much less noticeable.

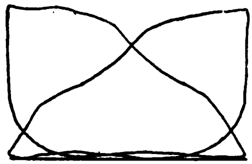
The hump also appears to approach the toe of the diagram; that is, the exhaust of the companion cylinder takes place earlier as the cut-off becomes earlier. This will be seen to take place if the Zeuner valve diagram be drawn for a Stephenson link motion. In the last diagram the hump is not visible, partly on account of the throttling due to the higher speed, causing the exhaust to be more gradual, and partly on account of the back pressure being greater throughout the whole of the stroke. The effect of the speed on the steam line is also marked, causing wire-drawing, which is indicated by the sloping steam line.

Diagrams taken from the engine when running light are given in Fig. 115. The steam is wire-drawn or throttled by the partial closing of the regulator valve.

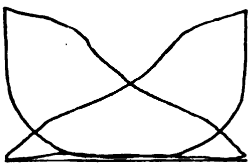
The hump in the exhaust line is visible in Fig. 114, but less marked in the diagrams in Fig. 116, which were taken from a locomotive under the conditions stated below each diagram. The range of load is greater than in the previous



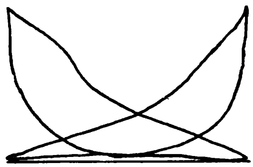
Notch 10, M.E.P. 114.
Cut off ·58 Stroke.
Speed 11 miles per hour.
I.H.P. 349.



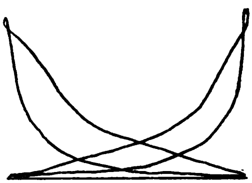
Notch 8, M.E.P. 97.
Cut off ·45 Stroke.
Speed 11 miles per hour.
I.H.P. 353.



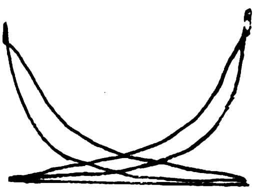
Notch 6, M.E.P. 74.
Cut off ·33 Stroke.
Speed 22 miles per hour.
I.H.P. 452.



Notch 4, M.E.P. 43·7.
Cut off ·23 Stroke.
Speed 39 miles per hour.
I.H.P. 473.



Notch 3, M.E.P. 26·7.
Cut-off ·15 Stroke.
Speed 40 miles per hour.
I.H.P. 296.



Notch 2, M.E.P. 19·6.
Cut-off ·15 Stroke.
Speed 69 miles per hour.
I.H.P. 376.

FIG. 116.

figure, and represents fairly well ordinary express work. The regulator was full open throughout.

The increase of compression with the earlier cut-off is very noticeable, and it will be found that this must be so if the Zeuner valve diagram is drawn for the Stephenson gear. The effect of speed on the steam line is very marked though this is not altogether due to speed. "Linking

COMPARISON OF INDICATOR CARDS OF SINGLE-EXPANSION AND COMPOUND LOCOMOTIVES.

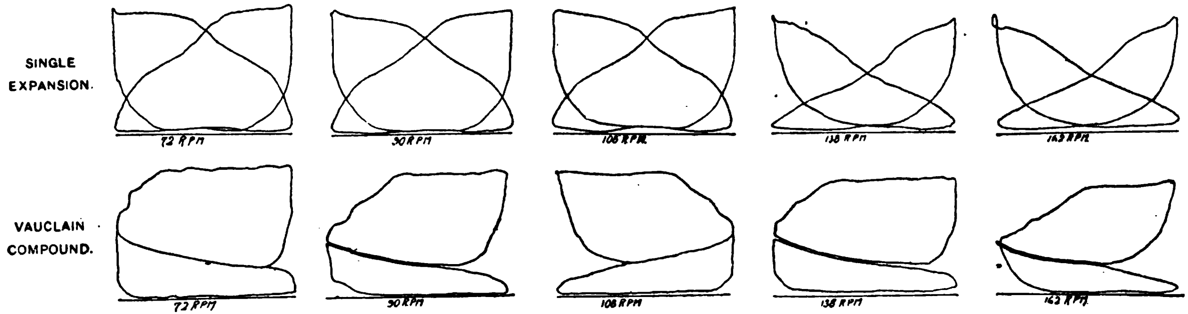


FIG. 117.

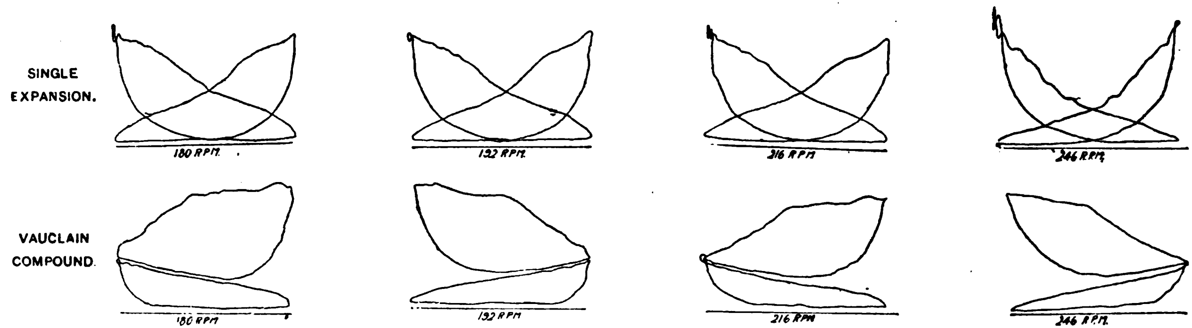


FIG. 118.—Diagrams taken from similar Baldwin locomotives, except that one was simple and the other compound.

up" causes the port opening to become less, and the less the opening the greater the amount of wire-drawing. So much is this the case that the point of cut-off becomes entirely obliterated on the diagram with much linking up, an instance of which will be found in the last pair of diagrams in Fig. 116.

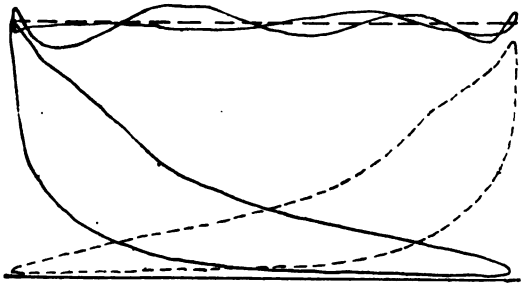


FIG. 119.—45 miles per hour.

The effect of speed upon the diagram is shown in Figs. 117 to 120, the reversing lever being half way between full and mid gear for the last two diagrams.

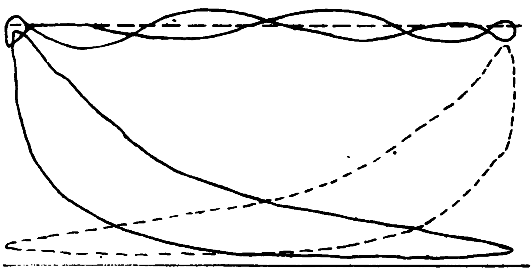


FIG. 120.—55 miles per hour.

The gradient was also as nearly as possible the same in both cases, though it is not recorded whether the speed was constant in both instances. The slower speed diagrams of Fig. 119 are rather fatter than those of Fig. 120, while the back pressure is greater at the higher speed.

Fig. 121 was taken with the regulator half open, the speed being 37 miles per hour. The full horizontal line is the boiler pressure line.

Friction Diagrams.—The diagrams in Fig. 122 were taken from a compound Corliss engine when running light, without any ropes on. The horizontal full lines are

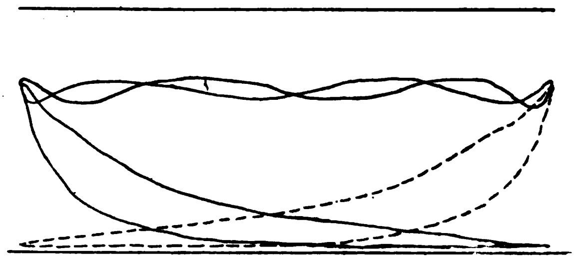


FIG. 121.—Regulator Half Open.

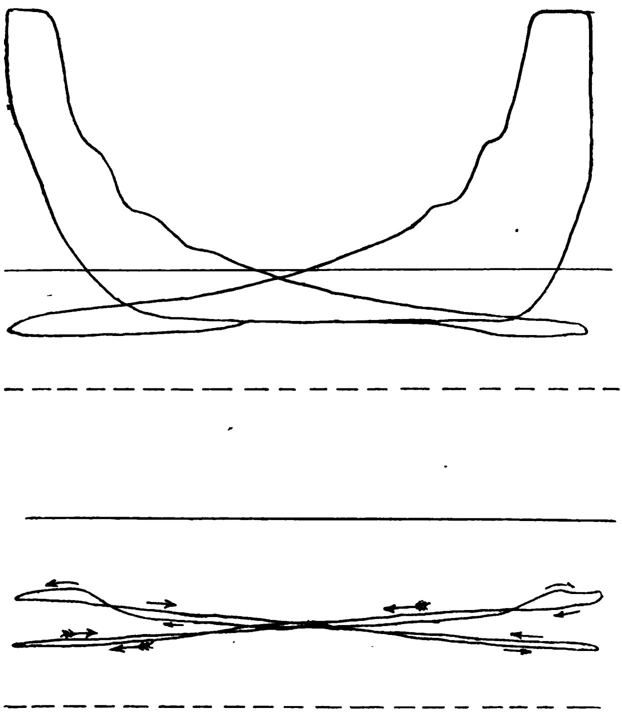


FIG. 122.—Friction Diagrams of Compound Corliss Engine.

the atmospheric lines, and those dotted are the absolute zero pressure lines. In both pairs of diagrams the pencil

was in contact with the paper for three minutes, so as to obtain a fair average diagram. The scale of the high-pressure spring was 16 and that of the low-pressure 10; the boiler pressure being 36 lb. per square inch.

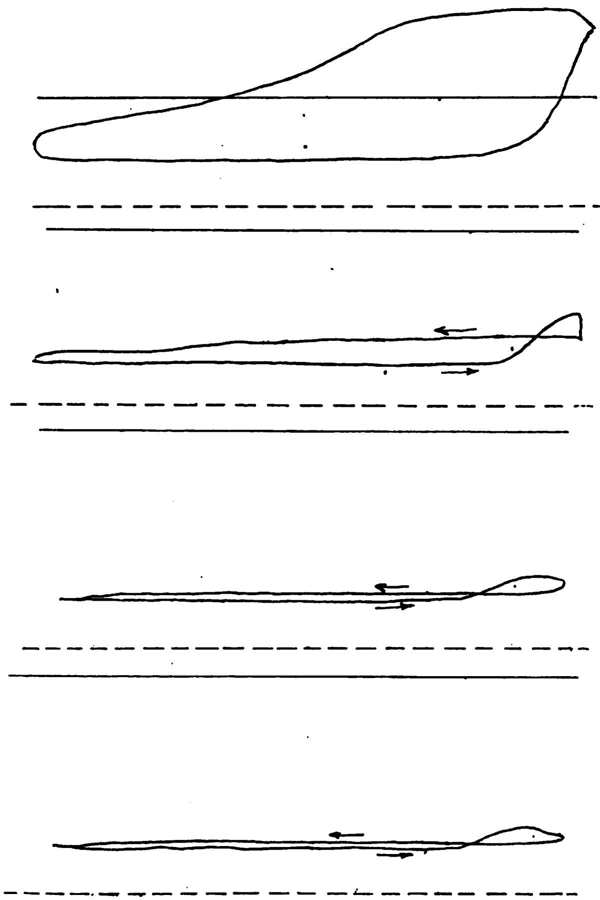


FIG. 123.—Engine Friction Diagrams.

The arrows in the low pressure diagram indicate the direction of motion of the pencil over the paper. The full economical load, for which the engine was designed, was 1,300 indicated horse power, the cylinders being 25 in. and 52 in. by 5 ft. stroke. The engine friction required 80·3

indicated horse power in the high-pressure cylinder and − 4·2 indicated horse power in the low-pressure cylinder, or a net friction horse power of 76·1.

If the mechanical efficiency were calculated on the assumption that the engine friction was the same at full as at light loads, then it would be

$$\frac{1300-76{\cdot}1}{1300} = {\cdot}94$$

but the friction is greater at heavy loads, though not necessarily much greater. The probable efficiency at full economical load would be about ·92, a very excellent performance.

Another set of friction diagrams taken from a four-cylinder triple-expansion Corliss engine are shown in Fig. 123. The cylinders were 25 in., 31 in., 42 in. and 42 in. diameter by 5 ft. stroke. The scales of the respective springs were 16, 10, 8, and 8. The absolute vacuum line is shown dotted in each diagram. The respective indicated horse powers were 105·8, 26·9, 4·3, and 3·9; total 141. The full economical load was 1,600 indicated horse power.

CHAPTER VIII.

THE ADMISSION AND STEAM LINES OF THE INDICATOR DIAGRAM.

The Admission and Steam Lines.—The character of the admission and steam lines depends upon a variety of circumstances, including amount of compression, size of ports,

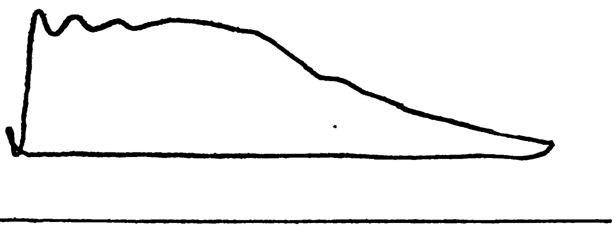


FIG. 124.—Eccentric Slipped Round on Shaft, causing Late Admission.

position of the eccentric which drives the valve, size of valve chest, length and diameter of steam pipe, and the speed at which the engine is running.

Typical examples of admission and steam lines will be found in Fig. 92, page 88, from a Corliss engine, and in Fig. 93 from an engine fitted with an ordinary slide valve.

FIG. 125.—Very Late Admission.

The actual admission line should be as nearly vertical as possible. The diagram in Fig. 124 was taken from the high-pressure cylinder of a compound engine at 300 revolutions per minute. The compression line returns upon itself, and then drops down to the level of the back-pressure line

before it begins to rise again, showing that the steam port did not open at the usual time, and admit steam at the end of compression.

As no steam entered the cylinder as the piston began its forward stroke, the steam that was compressed in the cylinder began to expand, but owing to very slight leakage, and probably some little condensation on the

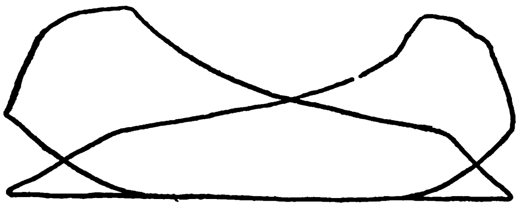


FIG. 126.—Late Admission.

walls of the cylinder, the expansion line is steeper than the compression line. When the forward pressure had reached the back pressure, the valve began to open the port to steam, and the admission line is sloping slightly forward. This slope is due to the opening being somewhat slow, and the difficulty the steam has in following up the retreating piston, and maintaining full pressure.

The eccentric had slipped backwards on the crank shaft, causing the different events to occur later than they ought to have done.

FIG. 127.—Late Admission.

Another and more marked case of late admission is shown in Fig. 125. If the valve and piston were quite tight, and there were no appreciable condensation, there would be no loop at the end of the diagram, and the downward part of it would return along the compression line, until admission began.

The actual shape of the admission line may not always be the same, though perhaps in a great measure due to the same cause.

For instance, in Fig. 126 we have the right-hand end rounded and the left-hand end pointed; while in Fig. 127 it is actually a cusp.

It is a question of the relative influences of a number of causes.

In both Figs. 126 and 127 there was too much outside lap on the valve at both ends, which gave no lead and very small port opening. The other features of the diagram are quite

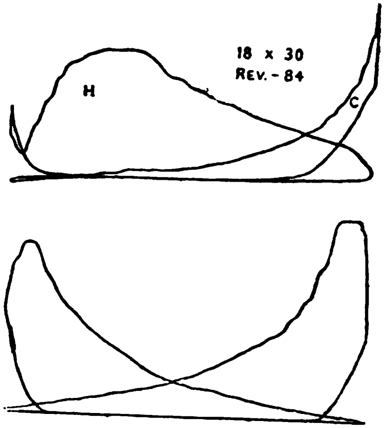


Fig. 128.—**Valve Rod of wrong length in upper diagrams, normal length in lower diagrams.**

regular, except that cut-off takes place earlier than it should with normal lap.

Fig. 128 shows late admission at the head end of the cylinder and correspondingly early admission and cut-off

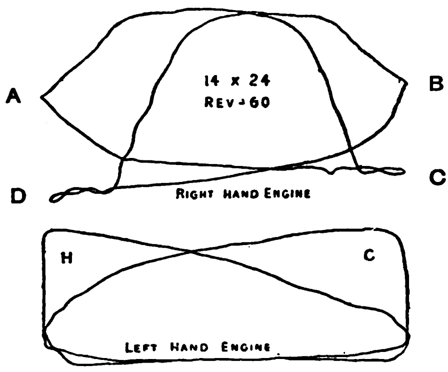


Fig. 129.

at the crank end, due to the eccentric rod being the wrong length. Normal diagrams are also given after the valve rod had been re-adjusted.

The diagrams in the next figure (129) were taken from a pair of high-pressure side-by-side engines, the lower ones being taken under normal conditions of working. The upper diagrams, it was stated, were due to the eccentric having slipped round on the shaft, causing late admission, &c., but it is much more likely from their appearance that the indicator cord was attached to the valve spindle or to a wrongly-arranged reducing mechanism. The crank end diagram begins at C, admission taking place after the stroke has commenced. Expansion begins near the end of the stroke and is apparently half completed at A. Similarly the other diagram begins at D, the point B being in the expansion curve. The reason for thinking the above diagrams to be due to defective reducing gear, rather than

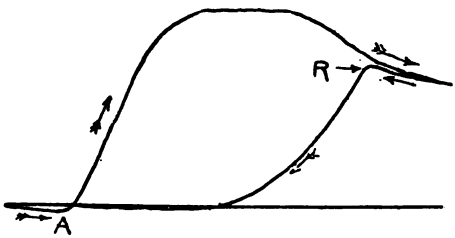


FIG. 130.—Eccentric set backwards.

a badly-set eccentric, will be evident on examination of the next figure (130) which is a diagram obtained when the eccentric was set much too far behind its proper position. Release takes place at R, and admission at A. There is no compression, and what expansion there is is almost useless. If the eccentric (Fig. 129) did slip backwards, and the reducing gear was correct, then the exhaust must have occurred immediately after A and B, with very slow opening of the exhaust ports. The author considers the defective reducing gear as the most likely explanation.

An instructive set of diagrams are given in Fig. 131. They were taken from the high-pressure cylinder of a three-stage compound engine, at intervals of twenty minutes, their order being indicated by the numbers attached to them. Steam was admitted to the inside edges of the piston valve of that cylinder. Just after the last diagram was taken, the top valve liner came out of its place. The gradual lifting of the liner caused a diminishing

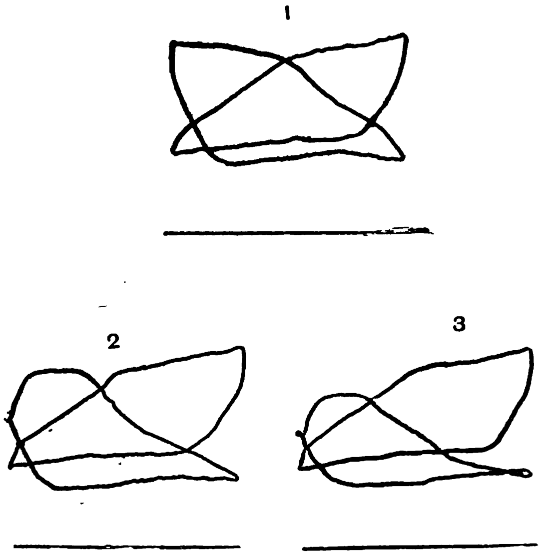


FIG. 131.—From High-pressure Cylinder of 3-stage Compound, 2 taken twenty minutes after 1, and 3 twenty minutes after 2, after which Liner came out of Valve Chest. Steam admitted inside. Admission got later as Liner lifted.

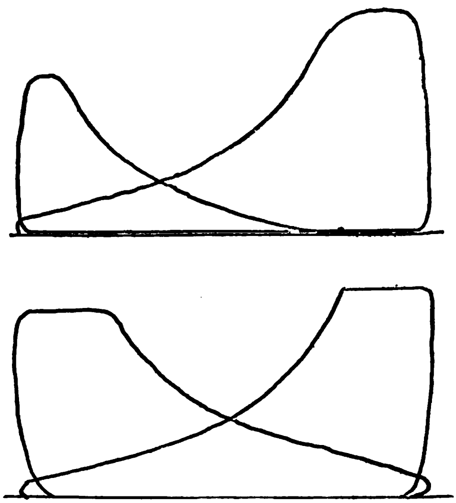


FIG. 132.

port area, and an increased lap on that edge of the valve, with earlier exhaust, and this is well shown in the left-hand diagrams of numbers 2 and 3.

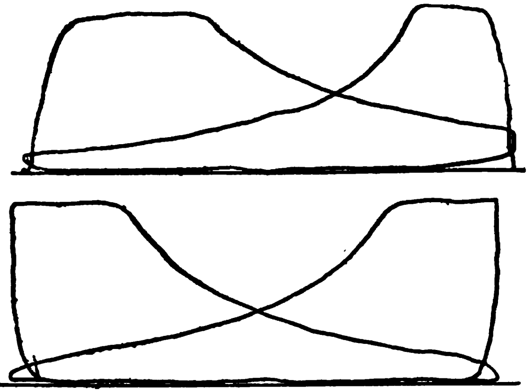


Fig. 133.—Eccentric Set Behind its Normal Position (Corliss engine).

The diagrams in Fig. 132 were taken from a Corliss engine, the upper ones, when the valves were wrongly set, giving no compression and late admission, while the port opening was much less for the head end than the crank

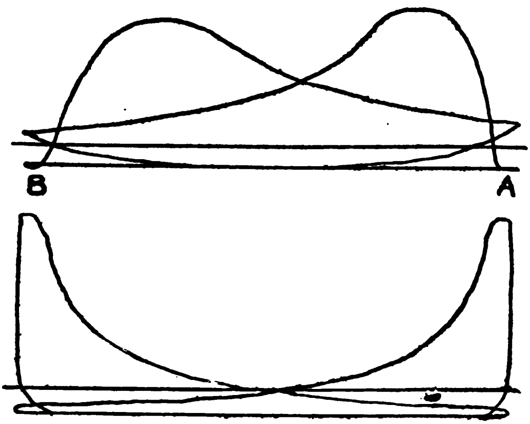


Fig. 134.

end, caused by too much lap. After a re-adjustment of the valves the lower diagrams were taken which show a much more even distribution of steam. The eccentric was advanced and the length of the eccentric rod adjusted.

In Fig. 133, the upper pair of diagrams show late admis-

sion with unequal cut-off at the two ends of the cylinder (Corliss engine). When the diagrams were taken the engine was pounding badly at each end of the stroke. The eccentric was advanced, after which the lower diagrams were obtained, showing good normal working.

A certain amount of compression seems to be necessary to prevent a large engine from pounding at the ends of the stroke.

In Fig. 134 a similar state of affairs prevailed, but worse in degree. The eccentric required advancing considerably, After readjustment the lower diagrams were taken.

Fig. 135 shows too early admission, the line being inclined backwards instead of vertical. This was due to the valve being displaced along the valve spindle beyond its normal position, giving increased lead.

FIG. 135.—Early Admission.

An instance of excessive advance of the eccentric (single valve engine) is shown in Fig. 136. Admission occurs too early by at least 30 per cent of the stroke, and

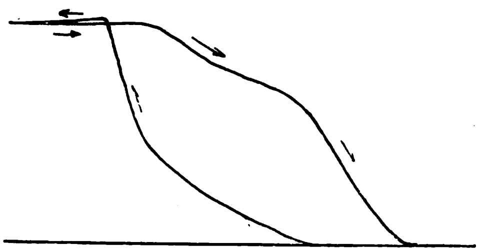


FIG. 136.—Excessive Advance of Eccentric.

the exhaust is about as much too early. Here the cut off was late. With an early cut off, it is possible to get the expansion line under the compression line.

Apparent excessive compression, causing a loop at the top of the diagrams, is shown in Fig. 137. These were taken from an engine fitted with a variable cut-off gear. The valve had $\frac{1}{16}$ in. lead in full gear, and 5 in. travel. It was running at time the diagrams were taken at 270 revolutions

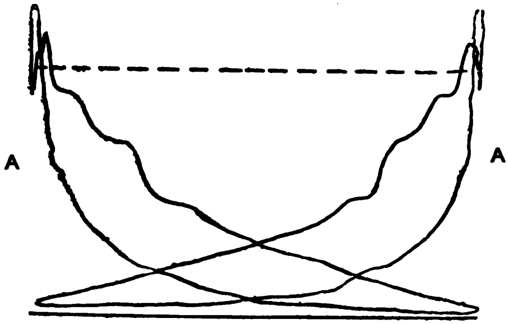


FIG. 137.—Apparent excessive Compression, but really early Admission.

per minute, and the boiler pressure (indicated by the dotted line) was 162 lb. per square inch by the gauge, while the tops of the diagram are 40 lb. above this. The loops are not due to compression, but to early admission, and the inertia of the moving parts of the indicator. Admission begins at A in both diagrams. (See also Fig. 139).

A very early as well as excessive compression is shown in Fig. 140, where a Trick valve was used. It was taken from the high-pressure cylinder of a compound locomotive.

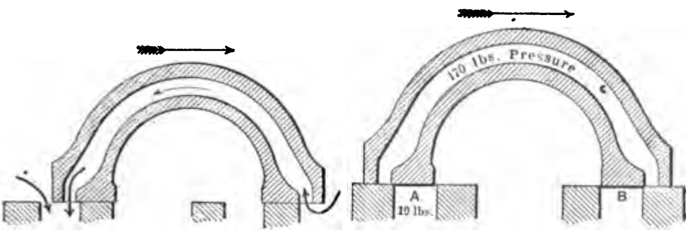


FIG. 138.—Section of an Allan-Trick Slide Valve.

In locomotive work, when the valve gear is notched up near the centre, and at high speeds, compression is almost always excessive, and a loop often occurs, though this can be diminished if sufficient port area be allowed, through which the exhaust steam can find an exit without the usual throttling.

Fig. 139 clearly indicates that the Allan-Trick valve is not necessarily responsible for excessive compression; but it is really due to too much lead being given to the valve. In Fig. 139 the arrows show the beginning of admission. The lead was a maximum in the uppermost diagrams and a

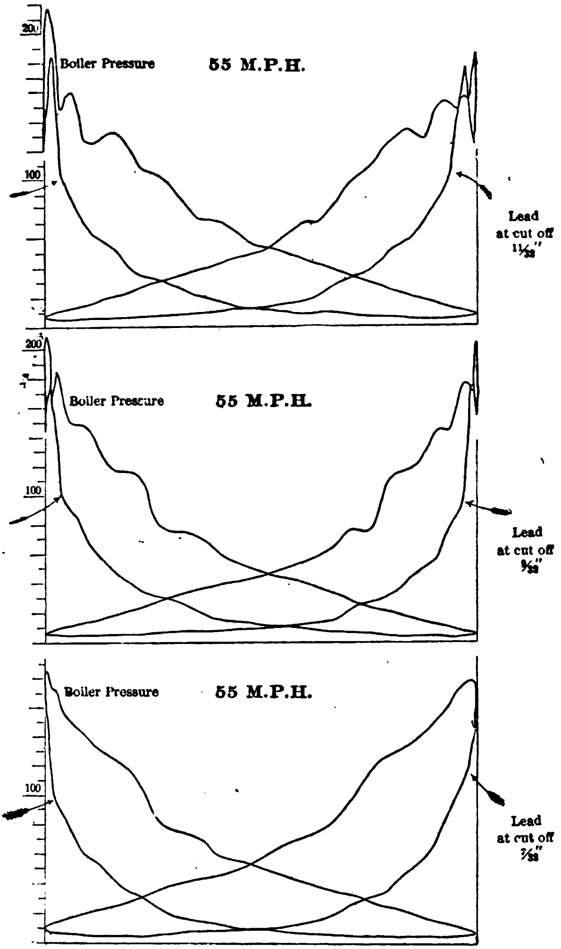


FIG. 139.—Diagrams from a locomotive fitted with an Allan-Trick valve.

minimum in the lowest diagrams. The wavy compression line is characteristic of the Allan valve, and is probably due to the steam imprisoned in the valve cavity being released into the cylinder during the early part of compression.

Figs. 141 and 142 contain diagrams taken from a locomotive running backwards, when the reversing lever was in the centre notch.

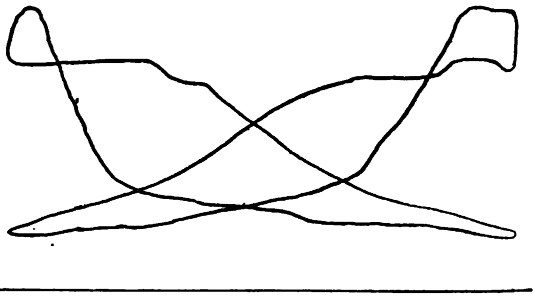


FIG. 140.—Taken from the high-pressure cylinder of a Compound Locomotive, cut off ·3 stroke, speed 12 miles per hour; boiler pressure, 135 lb. Allan-Trick slide valves.

It will be seen that compression begins at about mid stroke, and that admission begins sometime previous to the

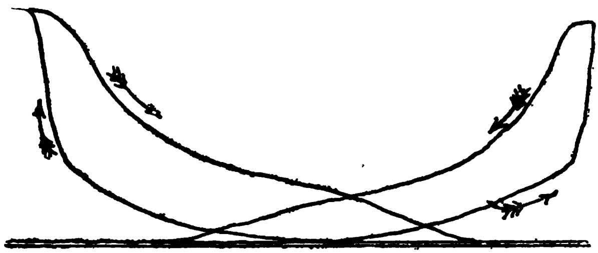


FIG. 141.—Running Backwards in Mid Gear Right Cylinder.

end of the stroke at one end of the cylinder, with the result that the steam line up to cut-off forms a sort of beak

FIG. 142.—Running Backwards in Mid Gear Left Cylinder.

to the diagram. Exhaust commences at about half stroke, and quite a considerable percentage is completed without

any pressure at all. This is only what we should expect with such excessive linking up. A Zeuner diagram shows it nicely.

When working at a high speed, or with restricted port openings, the parts of the steam line before and after cut-off

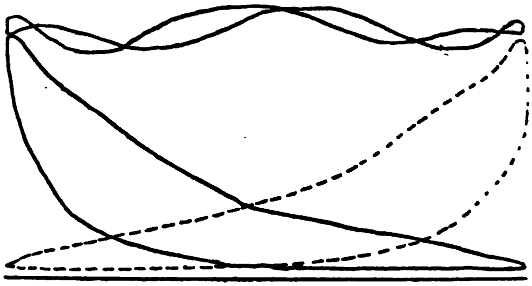


FIG. 143.—Speed, 48 miles per hour, Linked up three-quarters of Range from Full Gear.

often appear to be the same curve, or nearly so. This is especially the case in locomotive practice, and to a certain extent in marine practice.

An instance is given in Fig. 143, which was taken from a locomotive, and another in Fig. 144, which was taken from a four-cylinder triple-expansion marine engine working at about one-fifth maximum load, or one-fourth full economical load. A still better example is shown in Fig. 145, which was taken from the low-pressure cylinder of a compound locomotive, cut off ·42 stroke. Point of cut off entirely obliterated by wire drawing. A fourth example is given in Fig. 146, where the real points of cut-off are at A and B, the apparent expansion curves being due to very restricted port openings.

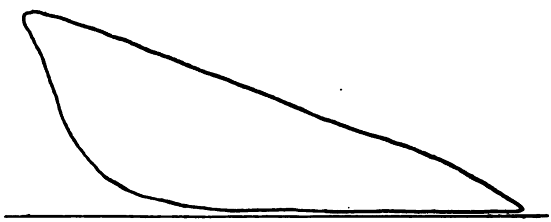


FIG. 145.

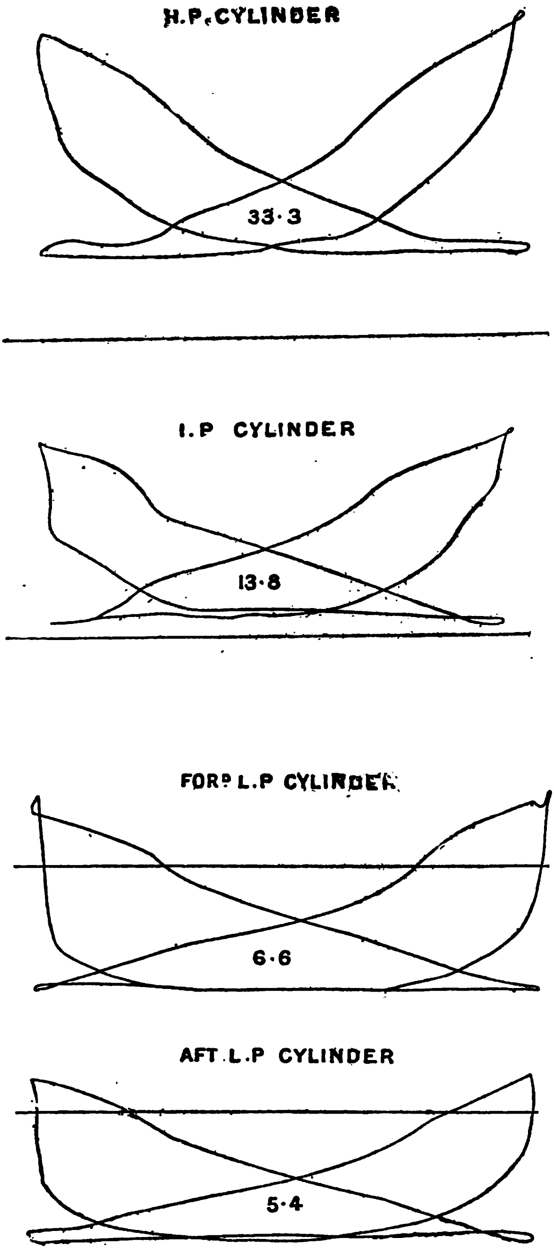

H.P. CYLINDER
33·3
I.P CYLINDER
13·8
FORD L.P CYLINDER
6·6
AFT L.P CYLINDER
5·4

FIG. 144.

The influence of a small steam pipe is well shown in Fig. 147, by the admission line of the steam chest diagram

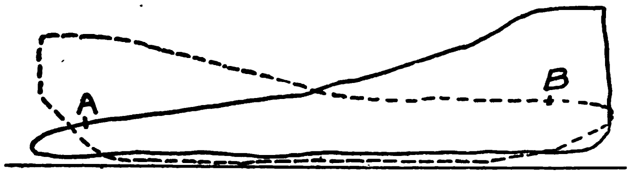


FIG. 146.—Restricted Port Openings. Speed, 60 revolutions. Real Points of Cut-off are A and B.

sloping considerably after the beginning of admission to the cylinder, and at the same time it approaches very

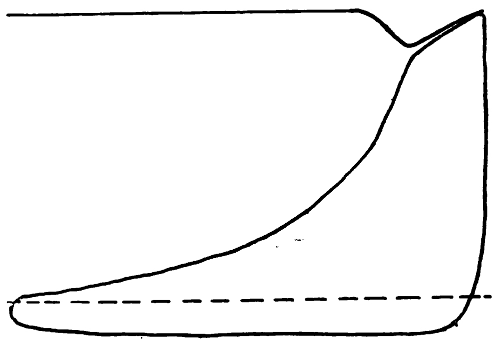


FIG. 147.

closely the steam line of the cylinder diagram. This shows a deficiency of steam in the steam pipe and separator if

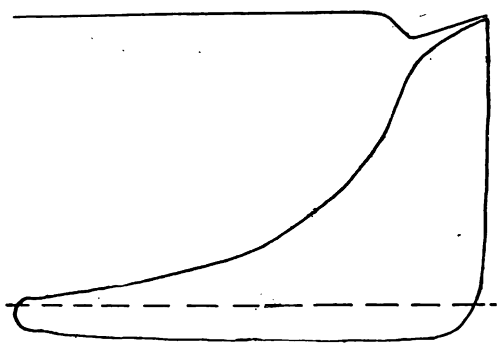


FIG. 148.

there is one, but as the steam chest steam line follows

closely the steam line of the cylinder, the steam has little difficulty it getting from one to the other, and hence the ports do not offer much resistance, that is, they are of ample proportions.

The next figure shows more port resistance, the two

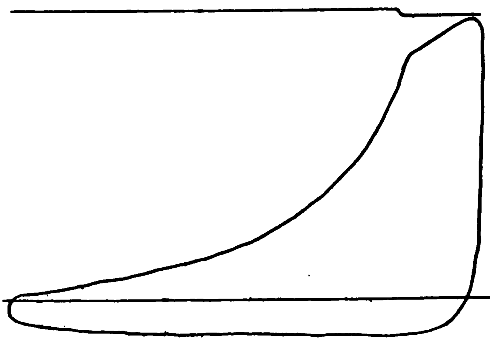


FIG. 149.

steam lines being separated further apart, while there is still too small a steam pipe.

Fig. 149 indicates that the steam pipe is large enough, but the port resistance is excessive.

The shape of the steam line is dependent somewhat on the receiver or steam-pipe volume. The dotted diagram in Fig. 150 was taken from the valve chest of a compound

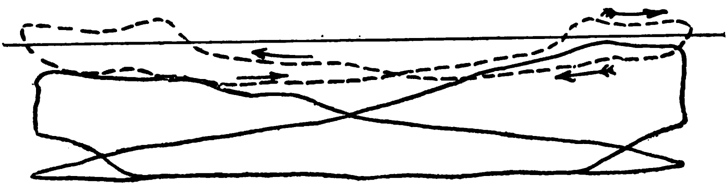


FIG. 150.—Dotted Diagram from L P. Steam Chest. No Receiver.

engine in which there was no receiver, and consequently a considerable fluctuation of pressure showed itself in the valve chest. There were 2 ft. of 5 in. pipe between the cylinders.

Throttling or wiredrawing, due to contracted port area, is shown in Fig. 151, the diagrams being taken from a horizontal tandem compound engine with an expansion slide valve in the high-pressure valve chest. The cylinders

were $23\frac{5}{8}$ in. and 44 in. by 5 ft. stroke, and the speed 60 revolutions per minute.

The diagrams are "combined," the high-pressure diagram being reduced in length in the ratio of the piston areas, for the purpose of comparing them with those which might be expected from an ideal engine.* We shall return to the method of combining diagrams in a future chapter, and

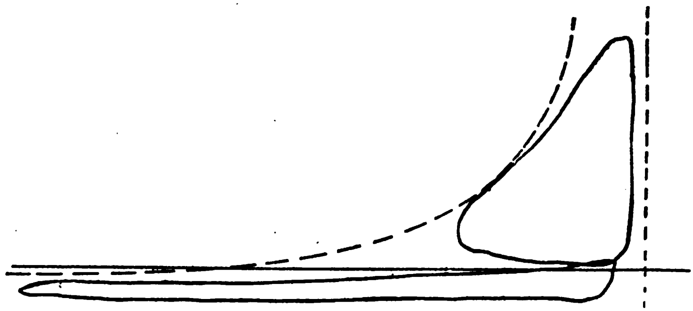


FIG. 151.—Much Loss of Area between Diagrams and Wire-drawing in High-pressure Steam Line due to Contracted Passages.

also to the application of the hyperbolic curve. All we shall state at present is, that the hyperbolic curve approximately coincides with the actual expansion curve in the best practice. (See page 181).

In Fig. 151 a considerable amount of throttling is shown in the steam line of the high-pressure cylinder due to the small opening permitted by the cut-off valve.

There may be some leakage into the high-pressure cylinder. There is further a lot of throttling between the cylinders, entirely obliterating the point of cut-off in the low-pressure cylinder.

The loss of area caused by wiredrawing is not so much a loss of heat, but the engine is not working up to its full capacity, and its mechanical efficiency is reduced. The best results are generally obtained without much wire-drawing.

It is difficult to distinguish between leakage into a cylinder and wiredrawing without testing the valve for tightness.

The diagrams in Fig. 152 were taken from a horizontal tandem compound engine, with cylinders 18 in. and 34 in. by 42 in. stroke, the speed being 60 revolutions per minute.

* For methods of combining diagrams, see page 175.

The cylinders were lagged and fitted with ordinary slide valves.

There is considerable wiredrawing in the high-pressure steam line, and also in the low-pressure steam line, due to restricted openings. There was no low-

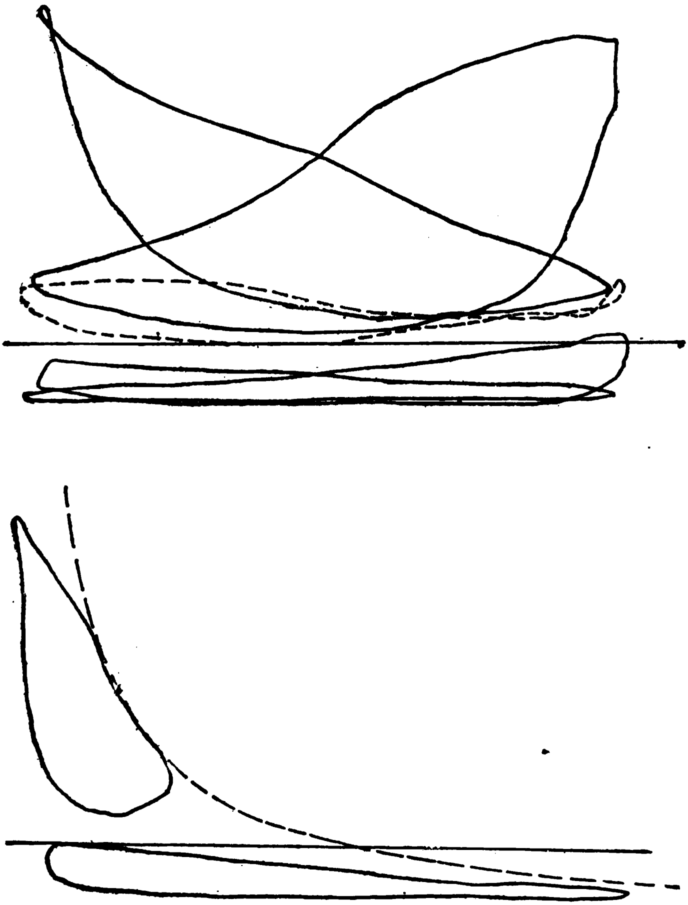


FIG. 152.—Horizontal Compound Tandem Engine. Cylinders, 18 in. and 34 in. by 42 in. stroke. Revolutions 60. Much Throttling in High-pressure Steam Line, and Low-pressure Ports.

pressure receiver, but 7 ft. of 6 in. pipe connected them together. The dotted diagram was taken from this pipe.

The gap between the cylinders is very large, and should not occur at so low a speed with properly designed ports and valve gear, though some drop in pressure must occur. An examination of the typical diagrams of economical working will give a fair idea as to how much is most suitable.

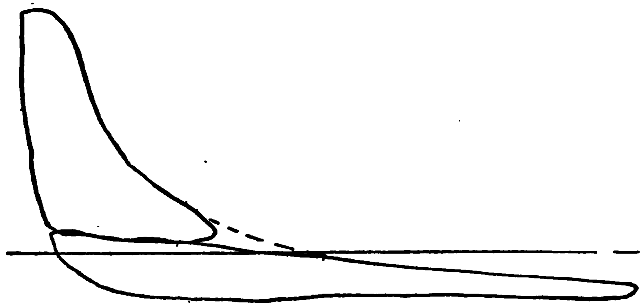


FIG. 153.—**Vertical Compound Side-by-side Engine. Cylinders 17 in. and 29 in. by 36 in. Revolutions, 70.**

The diagrams combined in Fig. 153 show no drop at all, and were taken from a vertical compound side-by-side engine, cylinders 17 in. and 29 in., by 3 ft. stroke, running at 70 revolutions per minute. The cylinders were lagged,

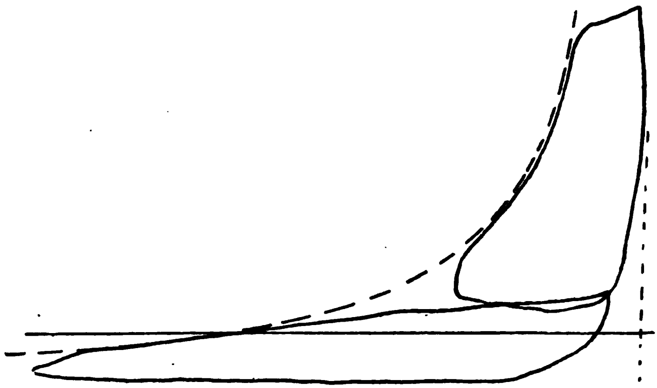


FIG. 154.—**Horizontal Tandem Corliss Engine. Cylinders, 17 in. and 30 in. by 5 ft. Stroke. Revolutions, 70.**

the high pressure being fitted with a cut-off valve. The two diagrams were not taken simultaneously, as the indicator had to be changed over from one cylinder to another, only one being available. A slight change of load

between the two indications will account for the absence of drop. The diagrams are individually very good. Though this method of indicating cannot always be avoided, it should never be resorted to whenever accurate results are required.

In Fig. 154 we have the same sort of thing, only in a more aggravated form; the steam line in the low-pressure diagram being higher than the exhaust of the high-pressure diagram. They were taken in this instance from a tandem engine whose high-pressure cylinder was fitted with Corliss valves.

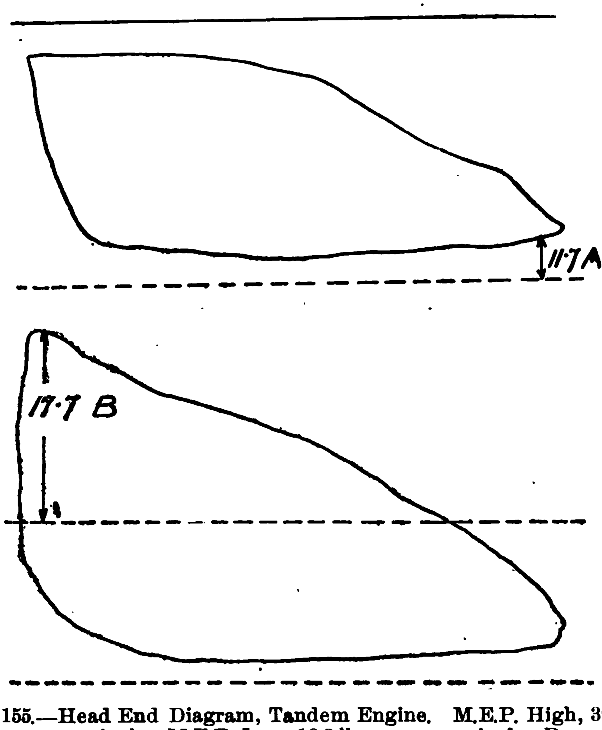


FIG. 155.—Head End Diagram, Tandem Engine. M.E.P. High, 39·2 lb. per square inch. M.E.P. Low, 18·3 lb. per square inch. Barometer, 14·93 lb. per square inch. Boiler pressure, 73 lb. Revolutions, 40 per minute.

The simultaneous indication of the cylinders of an engine prevents an unfair use being made of the indicator. In Figs. 155 and 156 the diagrams were taken from a tandem compound engine for the purpose of obtaining the indicated

horse power; but it was not possible to use more than one indicator. After the diagrams were taken and measured up it was discovered that the steam pressure in the low-pressure cylinder at A and B was in excess of the exhaust from the high-pressure at the corresponding points A and B. The engine was again indicated, but this time with a pair of indicators and no such result was then obtained. It was afterwards discovered that the engine attendant

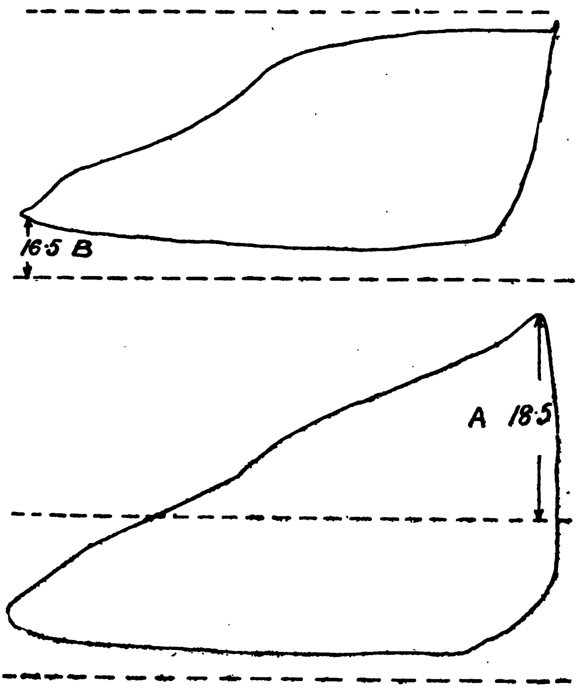


FIG. 156.—Crank End Diagram, Tandem Engine, High-pressure Cylinder, 22 in. diameter; Low-pressure Cylinder, 40 in. diameter; Stroke, 6 ft.; M.E.P. High, 39·4 lb.; M.E.P. Low, 17·9 lb.

had opened, with his foot, the bye-pass for admitting high-pressure steam into the low-pressure valve-chest at starting, the object being to make the engine appear more economical than it really was by developing a larger horse power on the same consumption of coal.

The diagrams shown in Fig. 157 were taken with a Tabor indicator from a Westinghouse compound engine at 280

revolutions per minute, the spring used being number 60. The drop there shown is about the normal quantity with this class of engine.

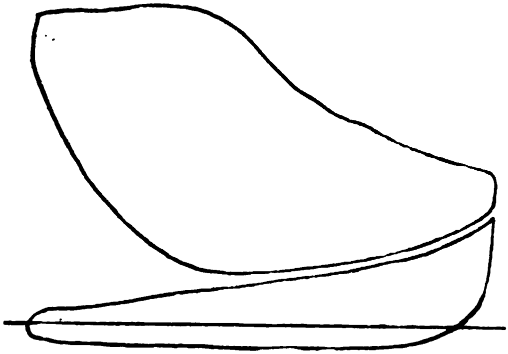


FIG. 157.—Diagram taken with a Tabor Indicator from a Westinghouse Compound Engine at 280 revolutions. Boiler pressure, 120 lb. Spring 60.

Another case of leakage into a cylinder is shown in Fig. 158. The cylinders were 8 in. and 14 in. by 16 in. stroke

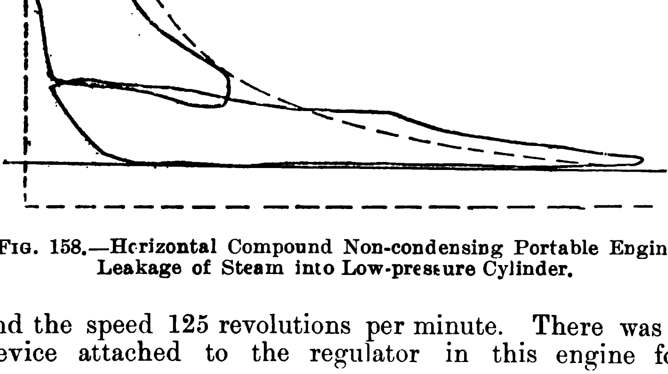


FIG. 158.—Horizontal Compound Non-condensing Portable Engine. Leakage of Steam into Low-pressure Cylinder.

and the speed 125 revolutions per minute. There was a device attached to the regulator in this engine for

allowing boiler steam to enter the low-pressure steam chest at starting, and there was a continuous leakage into the low-pressure valve chest while the engine was at work. The cut-off slide valve in the high-pressure cylinder did not open the port sufficiently to prevent the throttling in the high-pressure cylinder.

Figs. 159 and 160 present another instance of leakage. The former diagrams were obtained when the engine was working under normal conditions, while the latter set show how

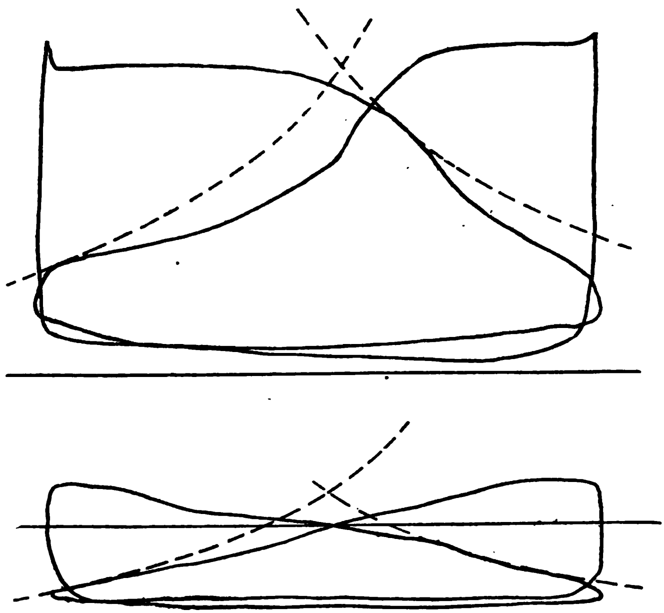


FIG. 159.—Horizontal Compound Engine. Cylinders, 20 in. and 36 in. by 54 in. Revolutions, 59 per minute. Diagrams Show Leakage. I.H.P. High, 202. Low, 110. Total, 312.

it was working when the cut-off valve in the high-pressure valve chest was out of order and not working. The steam consumption under the latter conditions was 13 per cent greater than when the cut-off valve gear was working. There appears to be an unequal cut-off in Fig. 159, with leakage into the high-pressure cylinder during one stroke and out of it during the other.

CHAPTER IX.

THE EXHAUST AND COMPRESSION LINES.

The Exhaust and Compression Lines.—The actual shape of the exhaust line is influenced by the cut-off, the speed and the relative dimensions of the valve and ports. Fig. 161 shows an exhaust line P Q taken from a small engine

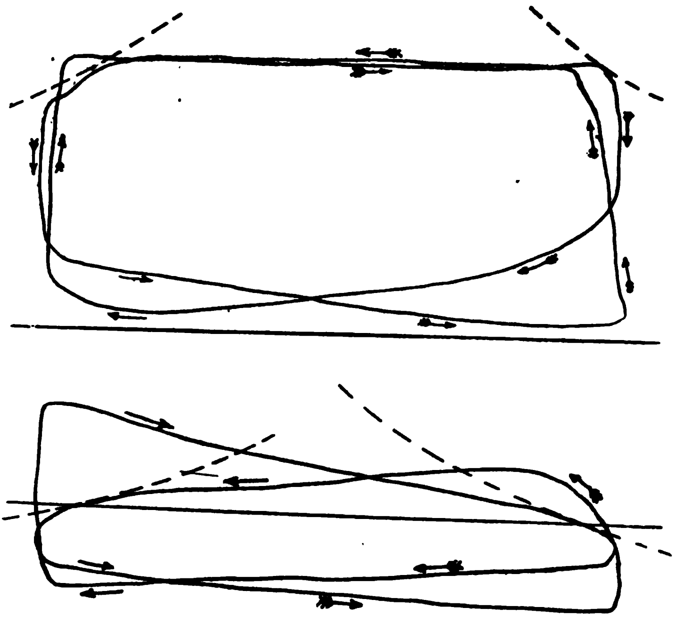


FIG. 160.—Diagrams taken from same Engine as those of Fig. 49, but after the Cut-off Gear was out of order and not working. I.H.P. High, 205; Low, 156. Total, 361. Revolutions, 56 per minute.

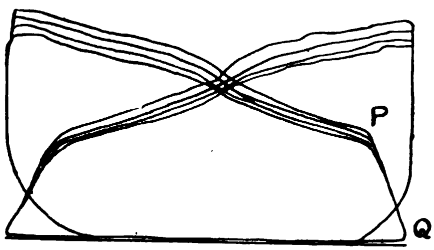


FIG. 161.—Speed, 200 revolutions per minute. Cut-off, ½ stroke.

running at 200 revolutions per minute, the ports being of ample dimensions; but when the eccentric is shifted across the shaft,* cutting off and exhausting earlier in the stroke,

* As in Fig. 195.

the exhaust line then appears like a miniature expansion curve, which is in a measure really what it is, the opening

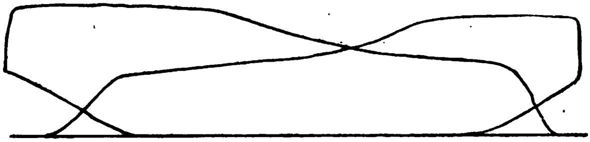


FIG. 162.—Speed "dead slow." Cut-off, $\frac{3}{8}$ stroke.

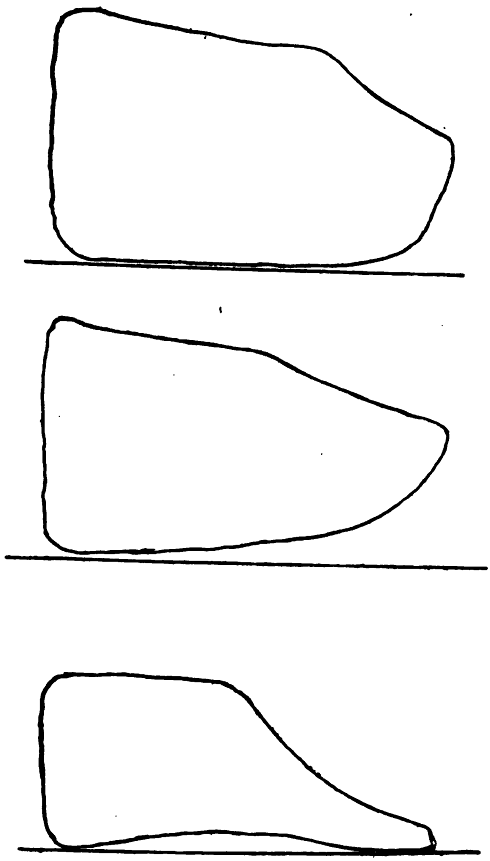


FIG. 163.—Faulty Exhaust Lines.

not being large enough to allow of the rapid egress of the steam.* Fig. 162 shows a diagram taken when the engine

* See also Figs. 93 to 95.

was running dead slow, and indicates that although the opening may be small, the piston is moving so slowly that the steam easily finds its way out of the cylinder before the piston has moved up to the end of the cylinder.

Late release is shown in the uppermost diagram of Fig. 163. Here the piston has returned along 25 per cent of the back stroke before complete release has been accom-

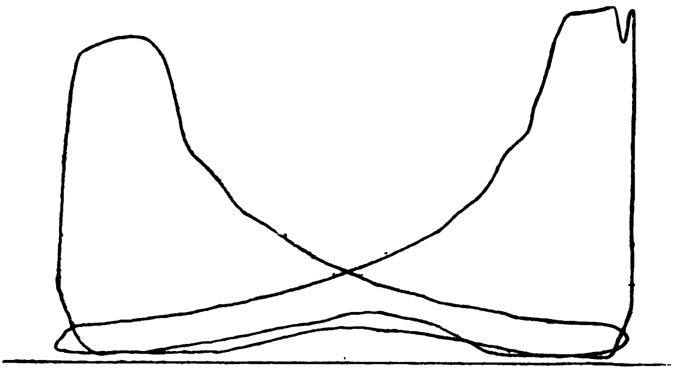


FIG. 164.

plished. The middle diagram indicates late release coupled with much throttling in the exhaust port, the opening being very small. The lowest diagram exhausts in good time, but on account of too much inside lap to the valve the opening was too much reduced and considerable throttling takes place while the piston is moving quickly near mid-stroke producing back pressure.

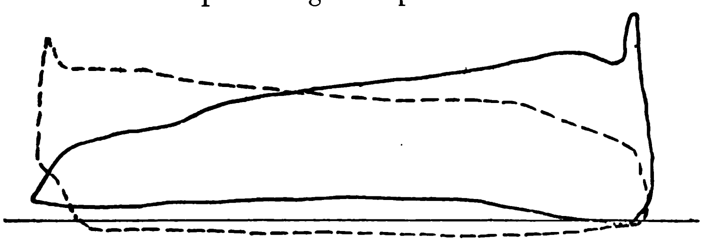


FIG. 165.

Something of a similar nature is indicated in Fig. 164, which contains a couple of diagrams taken from a non-condensing Corliss engine.

The movement of the exhaust valve, in the direction of opening, was too much, as it began to close the port

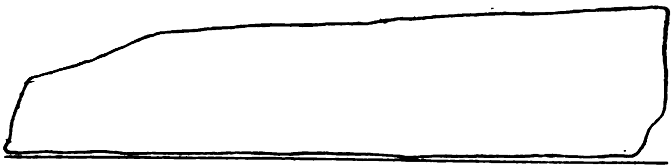


FIG. 166.—Revolutions, 19 per minute. Brake Load, 200 lb.

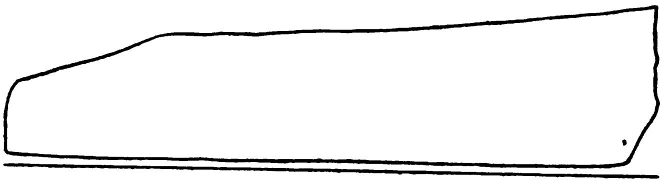


FIG. 167.—Revolutions, 36 per minute. Brake Load, 200 lb.

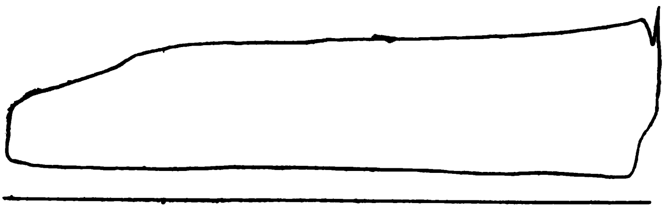


FIG. 168.—Revolutions, 66 per minute. Brake Load, 200 lb.

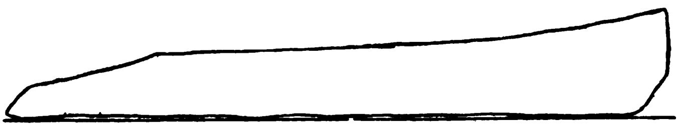


FIG. 169.—Revolutions, 30 per minute. Brake Load 100 lb.

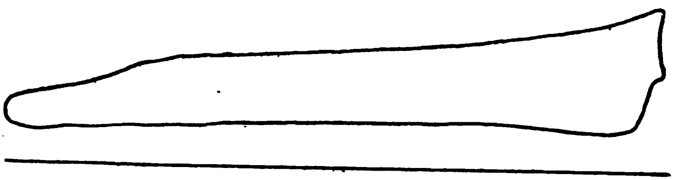


FIG. 170.—Revolutions, 96 per minute. Brake Load, 100 lb.

before commencing to return. This caused the rise in pressure in the exhaust line.

The high-pressure diagram from a compound engine is given in Fig. 165, in which the valve was not situated centrally with respect to the ports, the opening on one side being greater than on the other, causing throttling in one diagram, but not in the other.

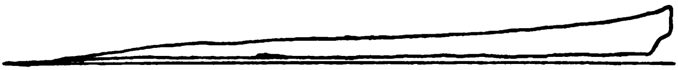


FIG. 171.—Revolutions, 62 per minute. No Load.

FIG. 172.—No Load. Exhaust much Crippled at the Higher Speed.

The next three figures show the influence of speed on back pressure, when the load is maintained constant. At the low speed of 19 revolutions the back pressure was small; at 36 revolutions it had not increased very much, but at 66 revolutions per minute it was excessive.

Two more diagrams taken at a different load are given in Figs. 169 and 170 from the same engine, in which the influence of speed is very marked.

The diagrams in Figs. 171 and 172 were taken at no load. It is well shown here that the back pressure is not due to the puff at the instant of release, but to the pushing out of the cylinder full of steam by the piston during the return stroke, the opening not being large enough for the steam to pass out through without considerable resistance.

The right-hand diagram of Fig. 173 presents a very odd appearance. It is only the exhaust line which is deranged, and it is due to the spindle of the exhaust valve being broken while it was covering the port, so that steam would be admitted as usual during the forward stroke, but would not be exhausted, and as the piston returned it would compress the steam again. The difference between the steam and compression lines is due to loss of pressure caused by leakage past the piston, or past the exhaust valve, or both.

The next pair of diagrams was taken from one side of a Vauclain compound locomotive, running at a speed of over 70 miles per hour. The high back pressure in the low-pressure cylinder, and the throttling between the

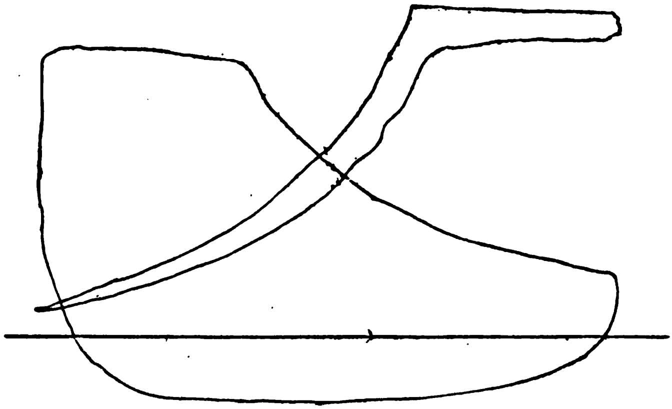


FIG. 173.—From a Corliss Engine Cylinder, one of the Exhaust Valve Spindles of which was broken.

cylinders, is due to the speed at which the engine was running, the piston speed being near 1,400 ft. per minute. The diagrams were taken with a Tabor indicator.

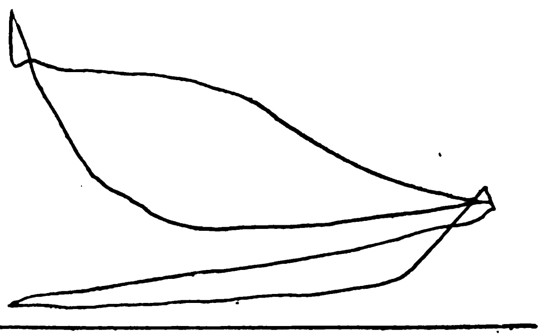


FIG. 174.—Taken with Tabor Indicator from Vauclain Compound Locomotive. Speed over 70 miles per hour.

The diagrams in Fig. 175 were also taken with a Tabor indicator from a locomotive at 7 miles per hour, working with 170 lb. per square inch boiler pressure. The peculiar humps in the exhaust lines are due to the puffs at

release from the other cylinder, temporarily increasing the back pressure. This is only noticeable with high release pressures, and disappears as the cut-off becomes earlier through linking up. This is very noticeable in the set of diagrams previously given on page 108.

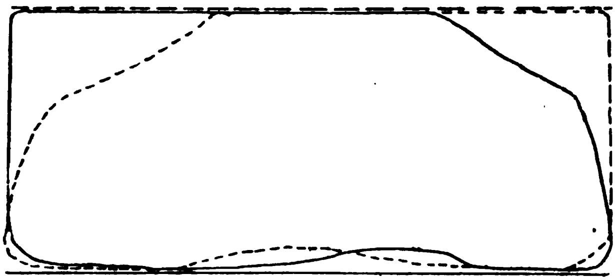


FIG. 175.

When running down hill with the regulator shut, the steam cylinders of a locomotive behave much in the same manner as pump cylinders for a portion of each stroke.

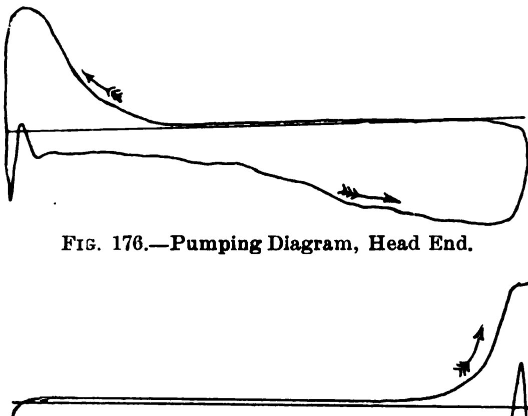


FIG. 176.—Pumping Diagram, Head End.

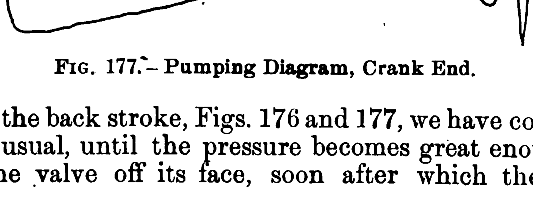


FIG. 177.—Pumping Diagram, Crank End.

During the back stroke, Figs. 176 and 177, we have compression as usual, until the pressure becomes great enough to force the valve off its face, soon after which the valve

opens the port to the valve chest and the piston begins the forward stroke, but without any steam to drive it; hence as the volume of steam chest *plus* cylinder expands, the pressure falls slowly till cut-off takes place, when the air in the cylinder expands and there is a more decided drop in pressure; in fact, we get the expansion line shown in the diagrams. When the exhaust port opens, air comes into the cylinder through the blast pipe and exhaust port,

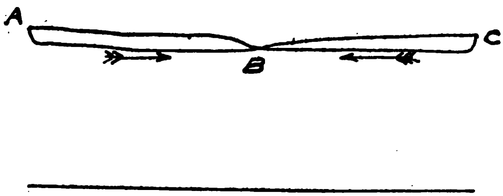


FIG. 178.

which raises the pressure to the back-pressure line. During the return of the piston the air which came in at the end of the previous stroke is pushed out again, until the valve closes the exhaust port and compression begins. In the above diagrams, the maximum vacuum was 9 lb. per square inch, and the maximum pressure above the atmosphere 10·5 lb. per square inch. The mean effective pressure would be about 6·5 lb. per square inch, acting in opposition to motion.

CHAPTER X.

VALVE CHEST AND STEAM PIPE DIAGRAMS.

Valve Chest Diagrams.—The few following diagrams were taken from the steam chest, and show the fluctuation of pressure that takes place there, which is in a measure an indication of the adequacy of steam-pipe sectional area. Fig. 178 is a diagram taken from a small vertical high-speed engine under full load at a speed of 350 revolutions per minute. The horizontal line below the diagram is the

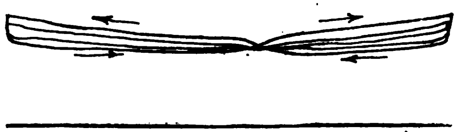


FIG. 179.—Showing Variation of Pressure in Steam Chest at Moderate Load.

atmospheric line. The steam was cut off at half stroke. As soon as the valve opened the port to steam the pressure dropped, as shown at A and C. At B the steam supply was cut off, and the pressure almost immediately rose to the boiler pressure. The length of steam-pipe was about 35 ft., and a separator was fitted next to the engine.

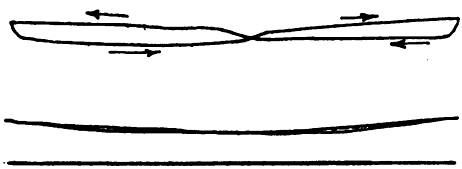


FIG. 180.—Steam Chest Diagram, at Full and no Load.

Fig. 179 shows how the steam pressure may be varied in the valve chest when working with a throttling governor. The load was light, and the speed 200 revolutions per minute.

In the next figure full and light load diagrams are given, at a speed of 350 revolutions per minute.

In Fig. 181 we have diagrams from a locomotive running at a speed of 48 miles per hour. Valve chest diagrams

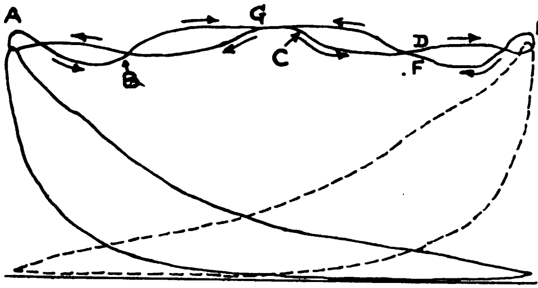


FIG. 181.—Steam Chest and Cylinder Diagrams from a Locomotive. Speed, 48 miles per hour.

from locomotives may vary to an enormous extent, and as the pipes connecting the valve chest to the indicator are not as a rule short, a considerable amount of oscillation often occurs.

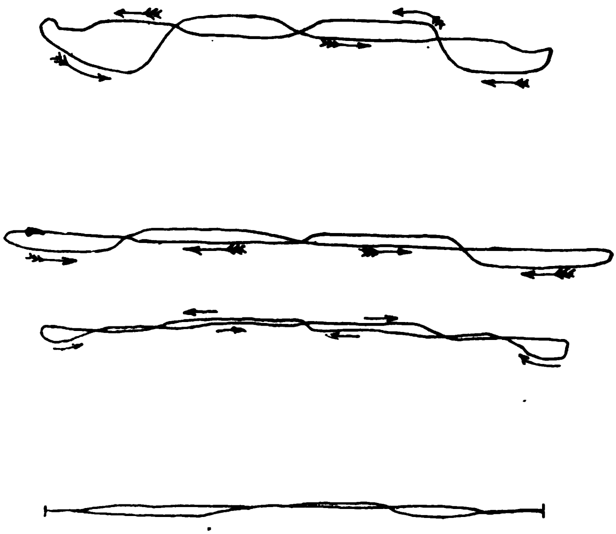


FIG. 182.—Steam Chest Diagrams.

Referring to the figure, the cylinder was commencing to take steam at A, which continued up to about B, after which the pressure rose to that of the boiler at G. There

is a certain amount of oscillation which rounds off the corners, and the length of the connecting pipe causes a certain amount of lag, so that the changes in pressure in the valve chest diagram appear to lag behind those which we should expect to occur.

At C the companion cylinder takes a charge causing a reduction of pressure. At E the first cylinder again takes steam, and the diagram is traced through F, and after cut-off the pressure rises to the boiler pressure, until at G the companion cylinder takes another charge.

Another set of steam chest diagrams is given in Fig. 182. The uppermost was taken from a Corliss engine running at 70 revolutions per minute, the cut-off taking place at 20 per cent of the stroke. The maximum fluctuation was 16 lb.

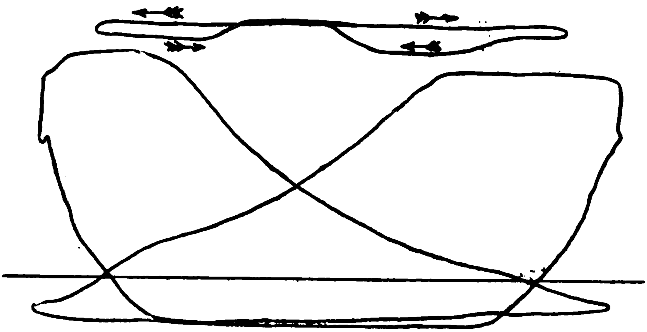


FIG. 183.

per square inch. The second diagram was taken from a similar engine at 60 revolutions per minute, but with a separator of inadequate volume. The third diagram was taken after a larger separator had been fixed. The last diagram was obtained from a compound Corliss engine, which was fitted with a separator of moderate capacity. The fluctuation is quite small.

The next diagram, Fig. 183, was taken from a McNaughted beam engine, the cylinder and steam chest diagrams being taken with different rigs, causing their lengths to be different. If the latter diagrams were expanded to the same length of stroke, it would be found that its features would agree very well with those of the cylinder diagrams.

In connection with the matter in this chapter the following figures in other parts of this work should also be studied: Figs. 85, 119, 120, 121, 143, 147, 148, 149, 150, 175, 197, 198, 202, and 203.

CHAPTER XI.

ADJUSTMENT OF THE SLIDE VALVE.

Valve Setting by the Indicator.—The state of adjustment of the valves in a steam engine can be much more readily and certainly ascertained by the use of the indicator than in any other manner. Many of the diagrams already given form exercises in valve setting, but those immediately following were taken during the adjustment of the slide valve, and consequently are more suitable for examples just here.

A small engine, which the author used, was fitted with a piston slide valve without rings. It was found that a considerable amount of leakage took place past the valve. It was then decided to try another piston valve in its place with packing rings, but which had not been made for the engine in question, though of the same diameter and approximately of the same dimensions elsewhere. The valve was put in place and the diagrams (Fig. 184) taken. As one diagram (head end) is very full while the other is hardly visible, it is evident there is a large opening at the head end and probably none at the crank end, the small diagram there shown being due to leakage. If the port did open, it would only be an infinitesimal amount. The valve was then raised about a quarter of an inch by adjusting a couple of nuts on the valve spindle. The diagrams Fig. 185 were then taken, but still the valve was too low. It was then raised another sixteenth of an inch, with the result shown in Fig. 186. After being raised a sixteenth more, the diagrams in Fig. 187 were taken, which show that so far as the length of the valve spindle is concerned the valve has been properly adjusted. But admission is late, and the cut-off, which was set at three-eighths of the stroke for the old valve, is much earlier with the new valve. This indicates that the valve opened late and closed early; in other words, there was too much outside lap on the valve. After removing the new valve, and comparing it with the old one, it was found to be a quarter of an inch longer than

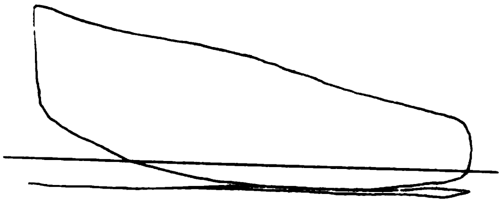


FIG. 184.

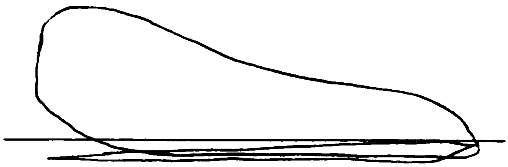


FIG. 185.

FIG. 186.

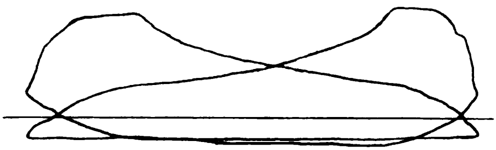


FIG. 187.

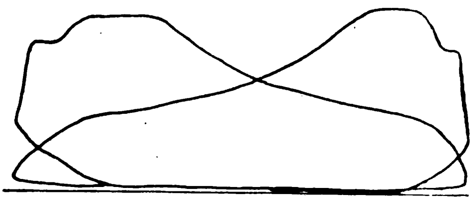


FIG.* 188.—Speed, 350.

* The peculiar shape of these diagrams at A and B is probably due to the sudden rush of steam at the beginning of admission, carrying the pencil much above its true position as in Fig. 106 ; and then being held there by friction until the steam pressure rose high enough to carry the pencil still higher. The true shape of the admission line would be a line joining the end of compression to the curve at the beginning of the horizontal part of the steam line. Compare this true admission line with those in Figs. 187, 126, and 127.

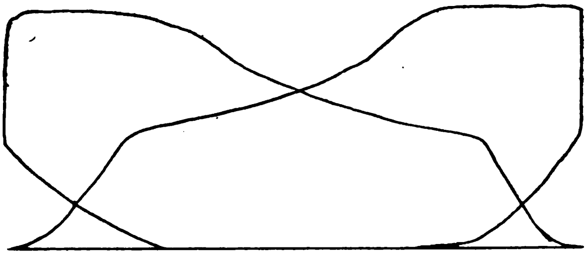


FIG. 189.—$\frac{3}{8}$ths cut-off. Speed, dead-slow.

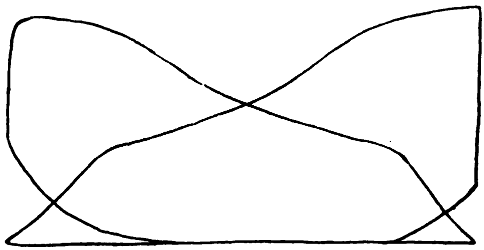


FIG. 190.—Speed, 163 revolutions.

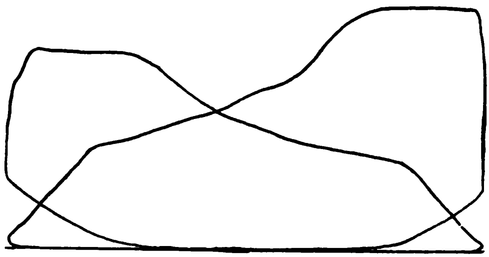


FIG. 191.—$\frac{3}{8}$ths cut-off. Speed, 212 revolutions. M.E.P., left, 42; right, 52 lb. per square inch.

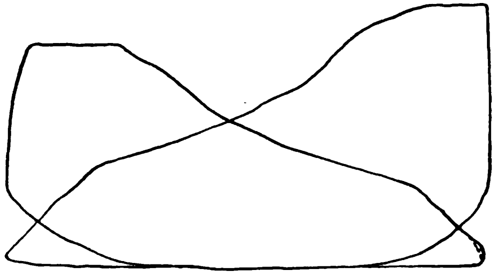
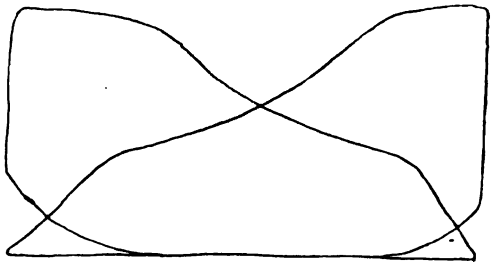


FIG. 192.—Speed, 288 revolutions

the latter. This was nearly all removed in the lathe, and Fig. 188 taken at 350 revolutions per minute. At this speed there was still evidence of late admission. A little more of the lap was removed, and Fig. 189 taken. The



Speed, 196 revolutions. M.E.P. in each cylinder 53·5 lb. per square inch.

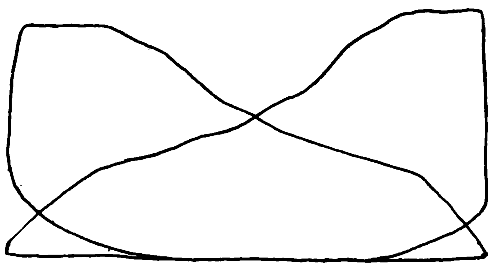


FIG. 193.—**Speed, 334 revolutions. M.E.P. in each cylinder 51 lb. per square inch.**

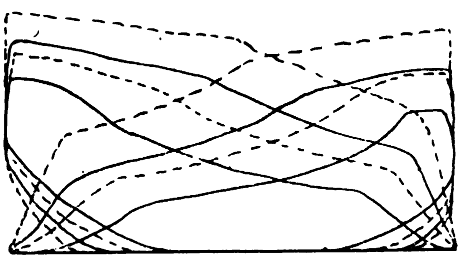


FIG. 194.—**Cut-off, $\frac{1}{4}$, $\frac{3}{8}$ths, $\frac{1}{2}$, and $\frac{5}{8}$ths. of stroke. Revolutions, 200. Steam, 137 lb. per square inch. Spring, 64.**

diagrams there are about equal in every detail, and it may be assumed that the valve was properly set. The speed at

which they were taken was only a few revolutions per minute. At 163 revolutions the diagrams Fig. 190 were obtained, which are quite satisfactory. At 212 revolutions per minute those in Fig. 191 were taken; and, further, those in Fig. 192 at 288 revolutions. While the cut-off is about the same in both strokes the opening at the head end must have been less than at the crank end, for on shortening the valve spindle, by removing a zinc liner from the eccentric rod, the diagrams in Fig. 193 were obtained at 196 and 334 revolutions per minute respectively.

The effect of a slight error in the length of the valve spindle is also more noticeable with an early than with a late cut-off. The diagrams in Fig. 194 were obtained when cutting off steam at $\frac{5}{8}$ths., $\frac{1}{2}$, $\frac{3}{8}$ths., and $\frac{1}{4}$ of the stroke, respectively, the whole being traced on the same sheet, and on the same atmospheric line after the diagrams were taken.

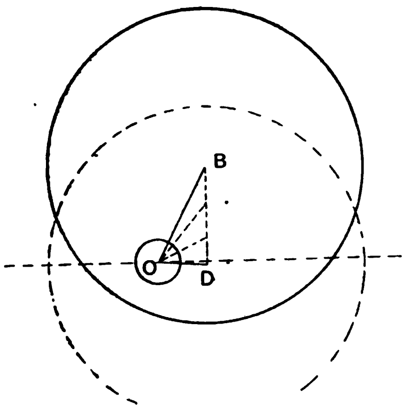


FIG. 195.

The speed was 200 revolutions per minute, and the boiler pressure 137 lb. per square inch. The stop valve was full open. At the latest cut-off, there is an indication that the valve spindle was slightly short, giving a little more opening to the head end than the crank end of the cylinder. But at $\frac{1}{4}$ cut-off there is an increase in the difference between them.

The alteration in the cut-off was effected by shifting the eccentric across the shaft, as shown in Fig. 195. O B is the position and length of the virtual eccentric arm, B being the centre of the eccentric at $\frac{5}{8}$ths. cut-off.

On one occasion the eccentric was moved till its centre was at D, just on the under side of the centre line, in which case it would just run backwards when once started, and the diagrams in Fig. 196 were taken. There was no load

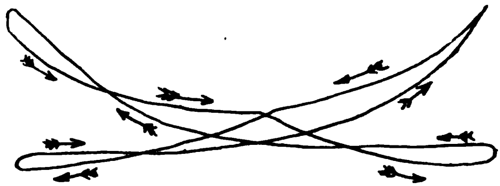


FIG. 196.—Running Backwards. Much Leakage. Running Light.

on at the time. The work of turning the shaft was done almost by the leakage past the valve, as it scarcely opened to steam with this position of the eccentric. It will be noticed that cut-off takes place almost simultaneously with admission, and exhaust begins at half stroke.

With Corliss engines the valves distribute the steam

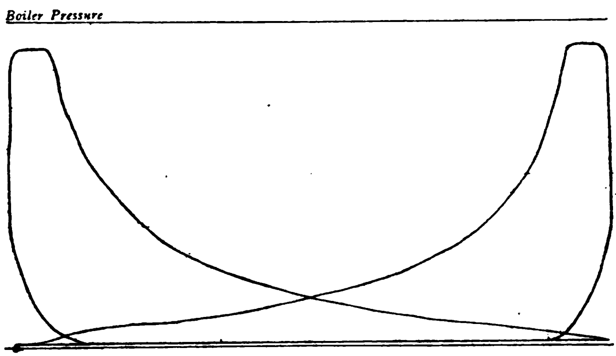


FIG. 197.—Engine running at 40 revolutions per minute and under ordinary conditions.

more perfectly than a single valve. Figs. 197 and 198 show very even distribution at both ends of the cylinder of a Corliss engine, with a very wide range in the cut-off, while Fig. 199 shows very unequal valve setting.

In Fig. 200 are two sets of diagrams taken from a locomotive, indicating bad valve setting. The eccentric rods were not of the right length, giving too much opening at one end and too little at the other. After resetting, the

diagrams in Fig. 201 were obtained, giving a more even distribution of steam.

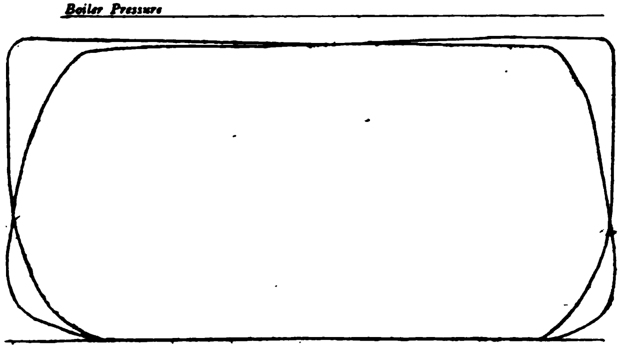


FIG. 198.—Engine running at 67 revolutions per minute under abnormally heavy load.

Fig. 202 shows two sets of diagrams, also taken from a locomotive, the upper set at 198 revolutions and cut-off at $\frac{5}{24}$ths. of the stroke, the regulator being full open. The valve travel was 6 in., the lap $1\frac{1}{2}$ in., $\frac{1}{8}$ in. inside clearance,

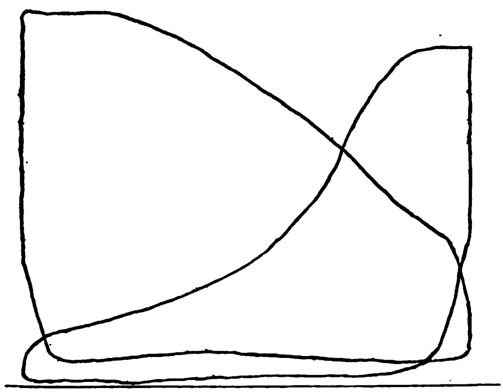


FIG. 199.—Diagrams from the high-pressure cylinder of old Marine Compound:
M E P top = 27·3 lb.
M E P bottom = 46·4 lb.
Very unequal valve setting, also much wiredrawing before cut off in left diagram.

no lead in full forward gear, and $\frac{1}{4}$ in. negative lead in full backward gear. A new valve having $\frac{1}{8}$ in. less outside lap, ^{3} in. inside clearance, $\frac{1}{32}$ in. lead in full forward gear, and

$\frac{1}{64}$ in. in full backward gear, was put in, and the second set of diagrams obtained with the regulator three-quarters open.

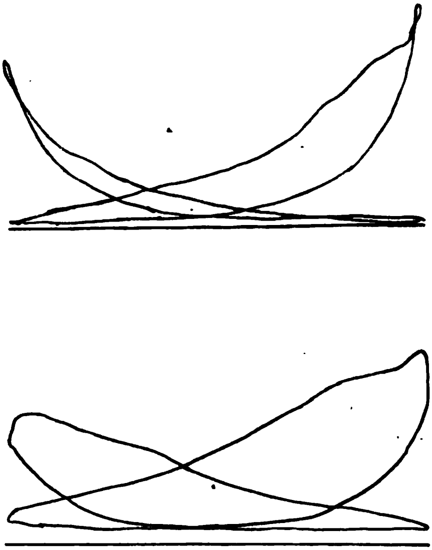


FIG. 200.—Locomotive Diagram, illustrating Bad Valve Setting. Upper Diagram, 1st notch; Lower Diagram, 3rd notch. Speed, 200 revolutions.

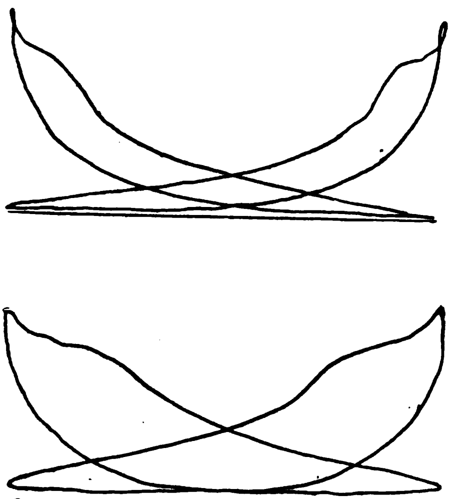


FIG. 201.—Diagrams taken after Valves in previous figure had been Re-set.

Three other sets from the same engine are given in Fig. 203. The first set was taken at starting, the M. E. P. being 166 lb. per square inch. The second set was taken at 88 revolutions, the cut-off being ·35 and M. E. P. 85 lb. per square inch; while the third set was obtained at 240 revolu-

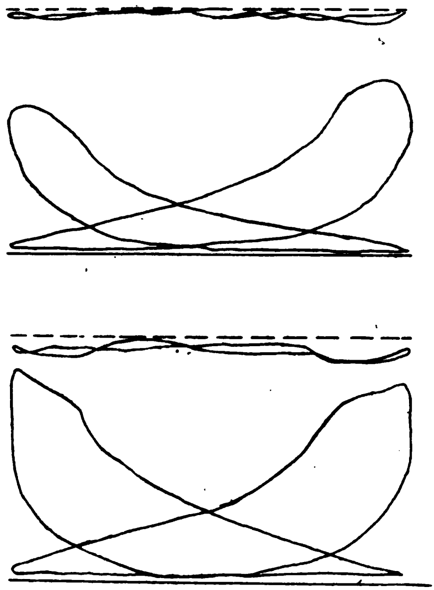


FIG. 202.—Top Diagram, 198 revolutions. Cut-off, $\frac{5}{24}$ths. Regulator Full Open. Valve, 6 in. Travel. Cylinders 19 in. × 24 in. $1\frac{1}{2}$ in. lap. $\frac{1}{8}$th inside clearance. No Lead in Full Forward Gear. $\frac{1}{4}$ Negative Lead Full Back.
Bottom Diagram. Same Engine. New Valve. 192 revolutions. $1\frac{3}{8}$ths lap; $\frac{3}{16}$ths in Clearance. $\frac{1}{32}$nd Lead Full Forward. $\frac{1}{64}$th Full Back. Regulator $\frac{3}{4}$ Open.

tions, with a cut-off of ·25 stroke and M. E. P. 49 lb. per square inch. The distribution is very even throughout.

Diagrams relating to valve setting might be multiplied almost without limit. Those already given serve to indicate the method of attack. As it is necessary to know full details as to the kind of engine, valve gear, speed, kind of indicator, and a host of other matters before some diagrams can be properly analysed, it is not proposed to go further into the steam engine diagram, with the exception of one or two examples of bad designs immediately following.

Meyer Expansion Valve.—A valve gear containing a separate expansion valve, working in conjunction with an ordinary distributing valve, will, when badly adjusted or designed, give a great variety of peculiar diagrams. Fig. 204

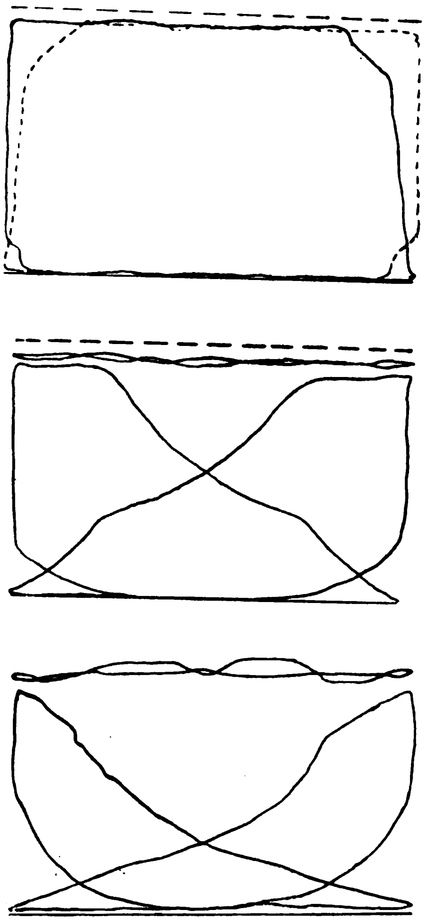


FIG. 203.—These 3 Sets of Diagrams were taken from same Locomotive as previous Diagrams. Fig .91.
1st at starting B P=180, M.E.P. 166.
2nd at 88 revolutions per minute. Cut-off, ·35. M.E.P. 85.
3rd at 240 revolutions per minute. Cut-off, ·25. M.E.P. 49.

was taken from an engine fitted with a Meyer gear, in which the cut-off plates overran the ports in the main valves

before cut-off by the latter took place, thereby permitting steam to enter the cylinder near the end of the stroke. The dotted toe was due to the same cause on another engine. Another instance of readmission is shown in

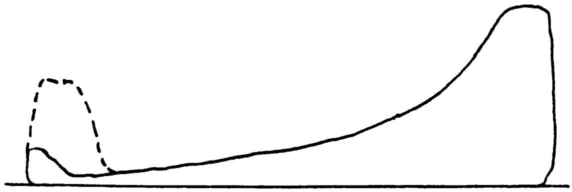


FIG. 204.—Showing Re-admission of Steam.

Fig. 205, where the cut-off plates were too narrow, and at the same time the indicator used was too heavy. The distributing valve was set with more lead on one side than the other, causing unequal compression and cut-off.

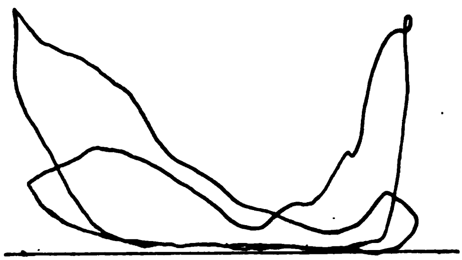


FIG. 205.—Re-admission with Gear Badly Designed.

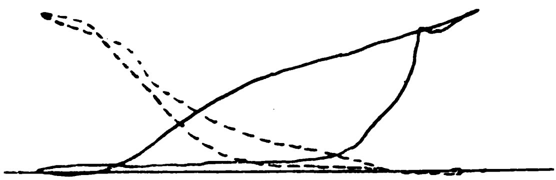


FIG. 206.—Meyer Gear. Distributing Valve has too much lead.

Fig. 206 is also from an engine fitted with a Meyer gear, in which the distributing valve eccentric is too much in advance of the crank, causing very early compression. The

next figure shows the opposite effect, the distributing valve eccentric not being set far enough in advance of the crank,

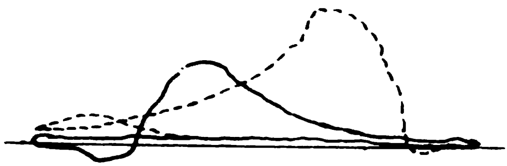


FIG. 207.—**Meyer Gear. Distributing Valve too late.**

with the result that steam does not enter the cylinder until the piston has moved through part of its stroke.

CHAPTER XII.

PUMP DIAGRAMS.

Water Pump Diagrams.—A diagram taken from one end of a waterworks' pump cylinder is given in Fig. 208. The speed is necessarily slow, and the head against which the water is being forced is constant.

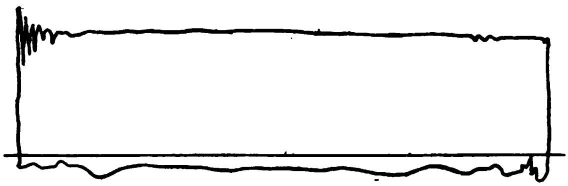


FIG. 208.—**Waterworks Pump Diagram.**

The next diagram, Fig. 209, was taken from a boiler feed pump. The temperature of the feed was not stated, but was probably high, as some of the stroke was occupied in compressing the vapour and air in the pump cylinder

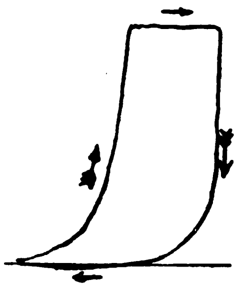


FIG. 209.—**Feed-pump Diagram.**

before any feed was forced into the feed pipe. The reverse is shown on the suction stroke. Only about half of each stroke is usefully employed.

Circulating Pump Diagrams.—Fig. 210 was taken from a circulating pump attached to a marine engine, in

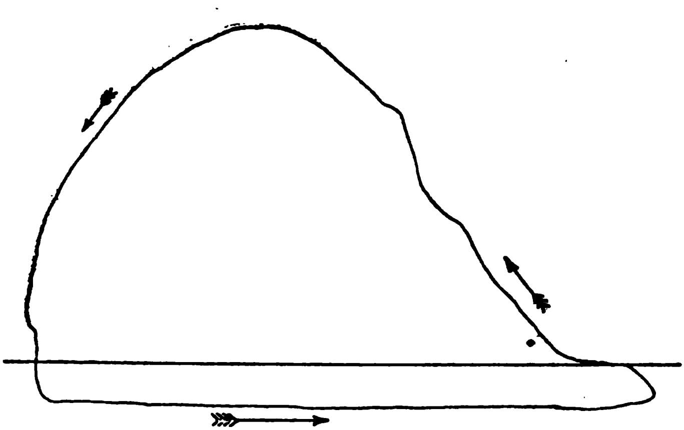


FIG. 210.—From Circulating Pump of a Marine Engine showing excessive pressure due to Cramped Valves and Passages. Maximum pressure 81 lb. above atmosphere.

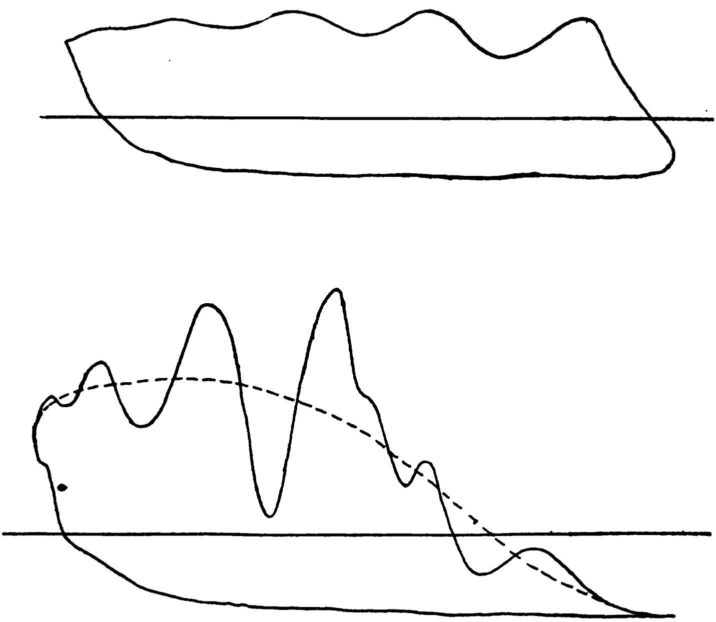


FIG. 211.—Diagram from Circulating Pump of Marine Engine, ample Valves and Passages. Top Diagram, 36 revolutions; lower, 59 revolutions.

which the delivery passages were very cramped, causing much resistance to the motion of the water.

The next pair of diagrams (Fig. 211) were also taken from a circulating pump, but in this case with ample passages.

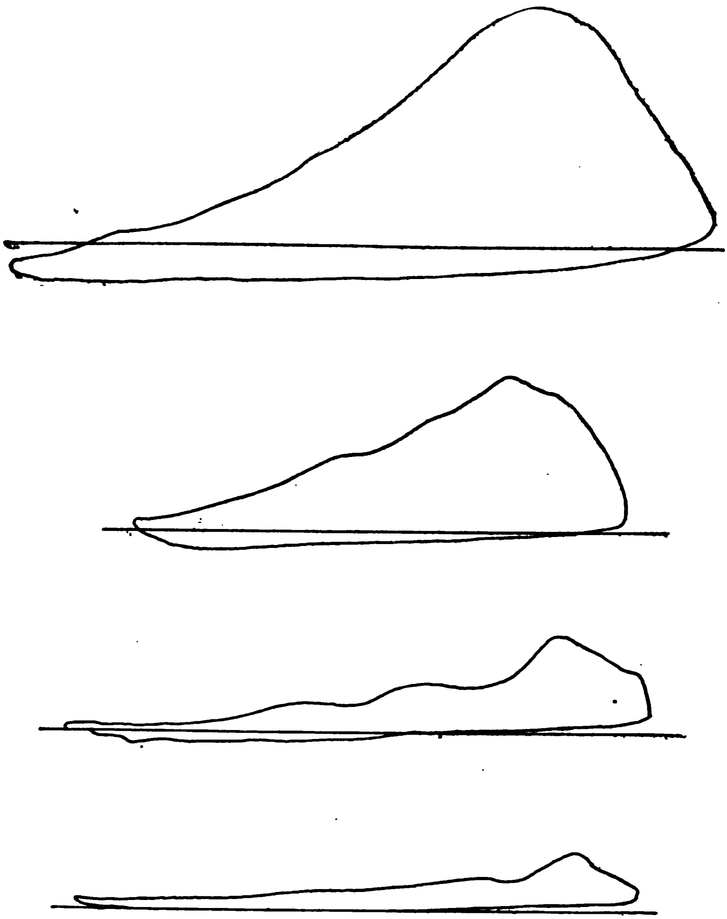


Fig. 212.—From badly-designed Circulating Pump (top), eventually replaced by better designed pump, from which following diagrams were taken.

At the slower speed of 36 revolutions the upper diagram was taken, the vibration there shown not being

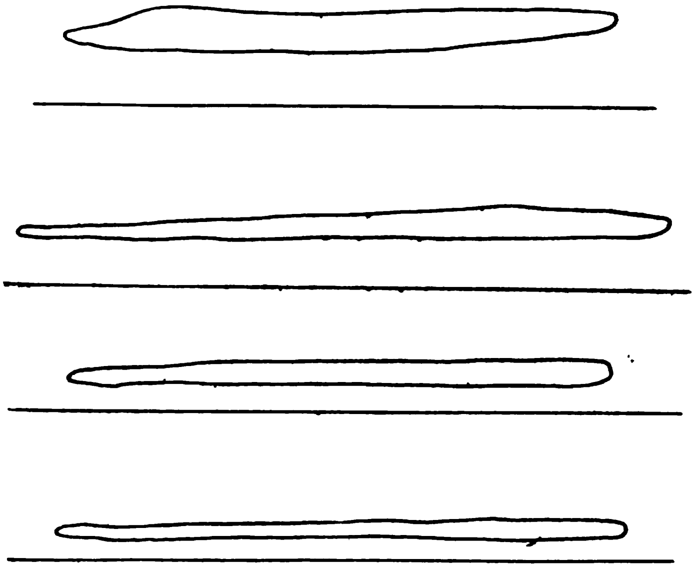


FIG. 213.—Diagrams from New Pump.

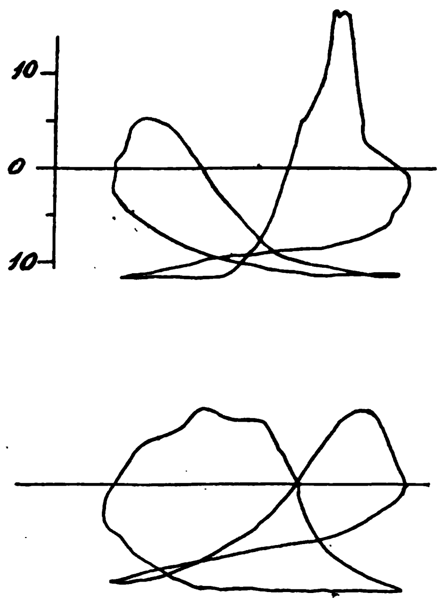


FIG. 214.—Jet Condenser, Vertical Air Pump Diagrams, double-acting, 84 strokes per minute, left-hand diagram from top of bucket.

excessive for a circulating pump. The lower diagram, which was taken at 59 revolutions per minute, shows a large amount of oscillation. This will generally be found in pumps working at anything but slow speed. The dotted line shows the mean pressure of the oscillations.

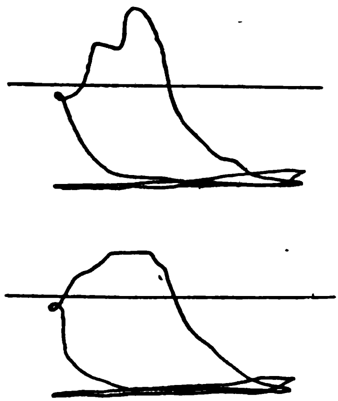


FIG. 215.—Same Air Pump working single-acting. Top without air valve, bottom diagram with air valve.

The four diagrams in Fig. 212 were taken from a badly-designed circulating pump, there being a valve in the pipe connection which could be used for throttling the circulating water. The uppermost diagram indicates the

FIG. 216.—Air Pump Diagram. Scale 10, Marine Engine, large valves and passages.

maximum amount of throttling by the height above the atmospheric line. This pump was replaced by a new one, and the diagrams in Fig. 213 were then taken to compare with those of Fig. 212. It must be borne in mind when comparing the two sets that all the diagrams in Fig.

212 were taken with a spring three times as stiff as those in Fig. 213.

Air Pump Diagrams.—Diagrams taken from a vertical double-acting air pump attached to a jet condenser are given in Fig. 214, the lower ones were taken after a snifting valve had been put on the pump for the purpose of getting rid of shock. The diagrams in the next figure were taken from the same pump, but when working single instead of double-acting.

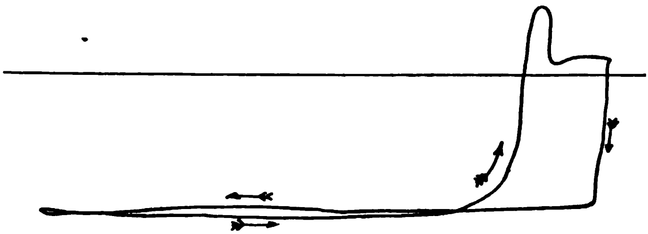


FIG. 217.—Diagram from Air Pump of Beam Pumping Engine.

Fig. 216 gives a diagram taken from a marine engine air pump with ample passages and valves, and Fig. 217 from a displacement air pump. The lower pressure on the compression stroke is not an uncommon occurrence, and is probably due to the longer time for condensing and probable

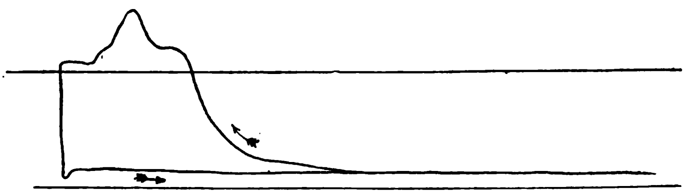


FIG. 218.—Diagram from one of Hudson's Displacement Air Pumps, made by Messrs. Hick, Hargreaves, and Co.

churning up of the water due to the motion of the displacing plunger causing more perfect condensation.

The diagram Fig. 218 was taken from one of Messrs. Hick, Hargreaves, and Co.'s displacement air pumps, shown in the next figure. The immediate production of a vacuum at the beginning of the suction stroke is very noticeable. A vertical and half transverse section of this pump are

given, showing all the valves at the top of the pump. This permits of the air and water coming freely into the pump chambers, while the air has not to come through the water,

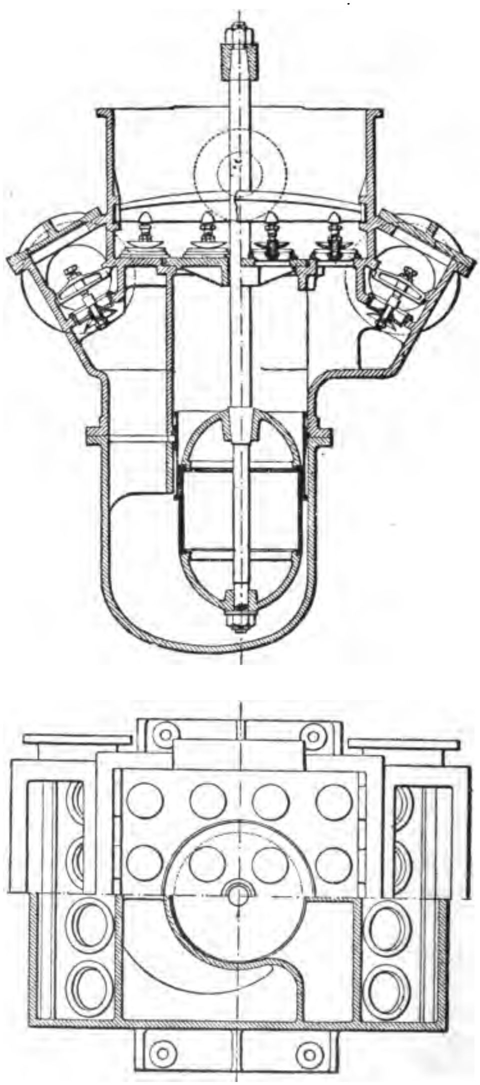


FIG. 219.—Hudson's Patent Displacement Air Pump.

as is the case with many air pumps. At the same time any clearance volume is filled with water, without this water pressing on the top side of any of the valves, thus getting rid of another objectionable feature in some pumps.

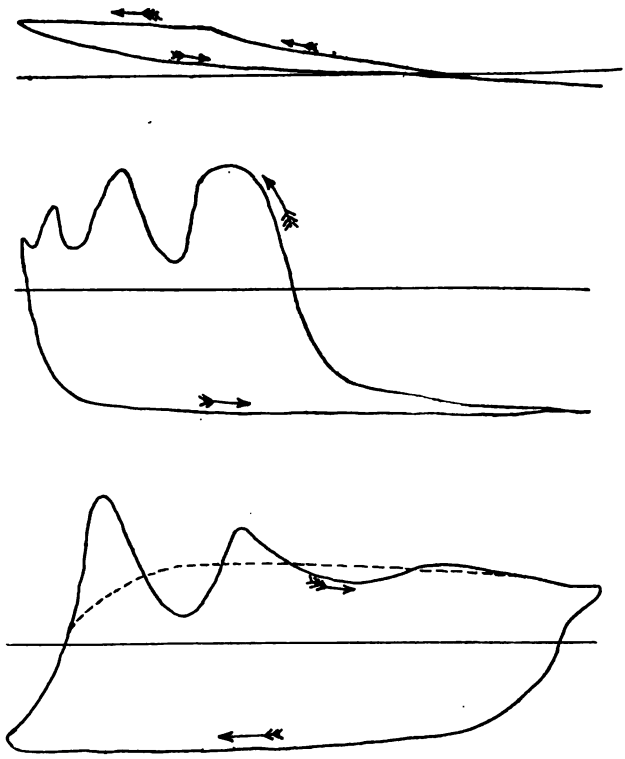


FIG. 220.—From Circulating Pump of Mill Engine, with various degrees of throttling in suction pipe, valves and passages ample.

1, top throttle almost shut; 2, top half open; 3, bottom full open.

The diagrams in Fig. 220 were taken from a circulating pump by the same makers, sections of which are given in the next figure.

Fig. 222 was taken from an air compressor of the displacement type.

One of the delivery valves was notched to allow water to run back and fill up the clearance space. The vertical lines in the middle of the suction stroke are due to the pump

pulling up until the leakage through the notched valve reduced the difference of pressure on the opposite sides of

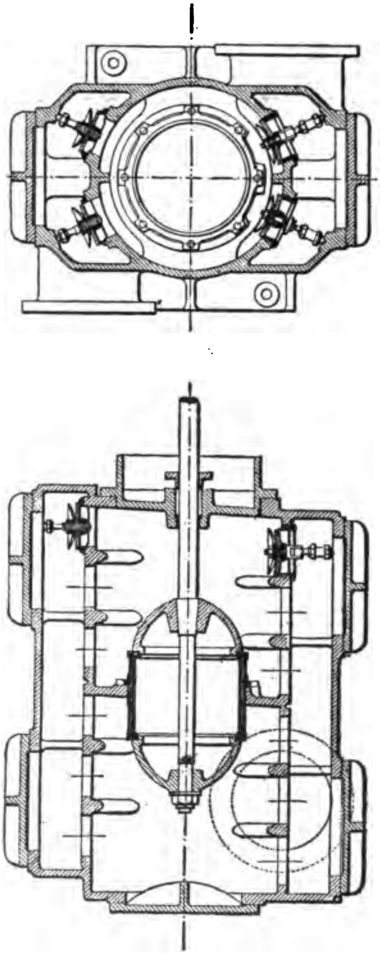


FIG. 221.—Circulating Pump.—Messrs. Hick, Hargreaves, and Co.

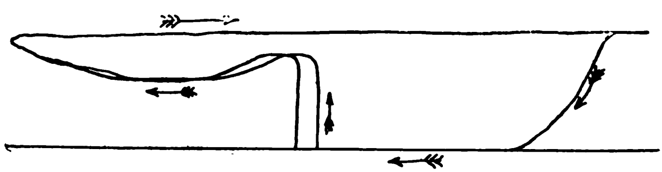


FIG. 222.

the piston, after which the stroke was completed. It was a double-acting compressor.

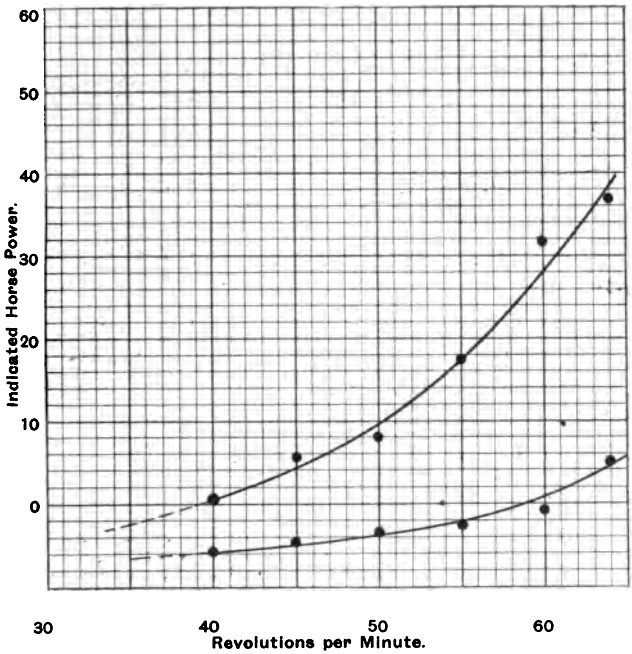


FIG. 223.

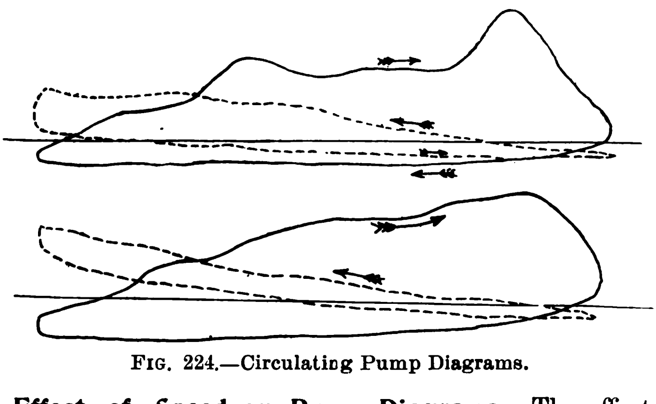


FIG. 224.—Circulating Pump Diagrams.

Effect of Speed on Pump Diagrams.—The effect of speed on the power at which the engine must work to drive a pump is well shown in the accompanying Fig. 223.

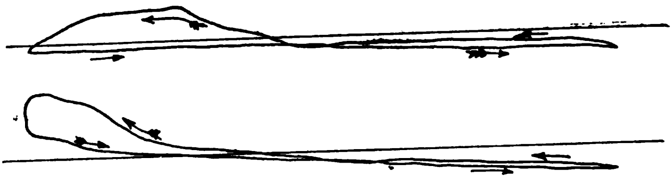


FIG. 225.—Bilge Pump Diagrams.

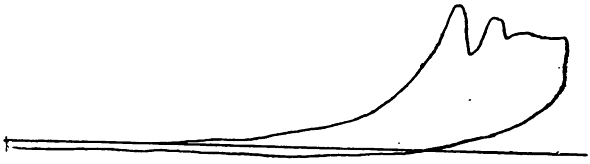


FIG. 226.—Feed Pump Diagram. Speed, 40 revs. per minute.

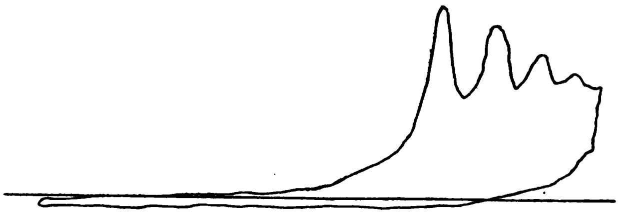


FIG. 227.—Feed Pump Diagram. Speed, 60 revs. per minute.

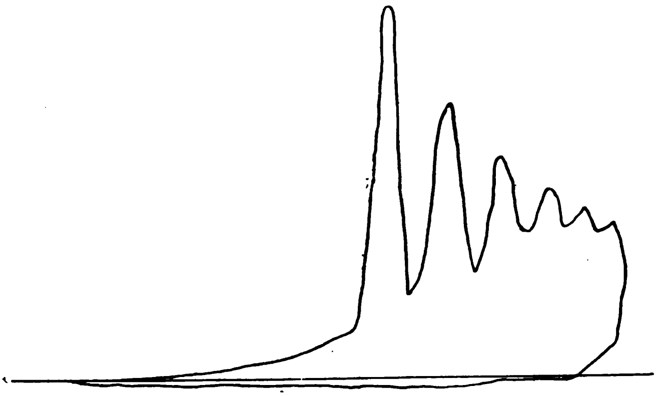


FIG. 228.—Feed Pump Diagram. Speed, 64 revs. per minute.

The upper curve refers to a circulating pump and the lower one to a feed pump. The power increases rapidly with the speed.

A couple of sets of circulating pump diagrams are shown in Fig. 224, the former of which was taken at 64 revolutions per minute and the latter at 45 revolutions. The scale of the former is double that of the latter. A couple of bilge pump diagrams are shown in Fig. 225. Figs. 226, 227, and 228 are diagrams taken from a feed pump at different speeds.

CHAPTER XIII.

MISCELLANEOUS DIAGRAMS.

Gas-engine Diagrams.—Four consecutive diagrams, taken from a 5 brake horse power gas engine, are given in Fig. 229. These may be taken as representative of satisfactory working for a small engine without a timing valve. The instant of firing the charge varies slightly—the earlier ones giving the higher explosion pressures. The slower the

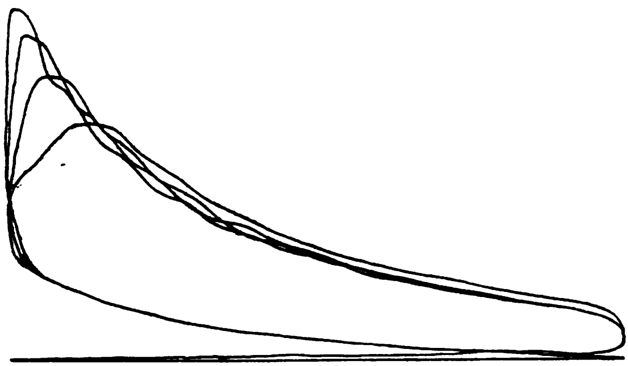


FIG. 229.—**Typical Gas Engine Diagram, nearly Full Load.**

ignition, the higher will be the expansion line towards the middle and end of the stroke, probably because the maximum temperature is less, and consequently less heat passes into the water jacket during the early part of the stroke.

In the upper part of the next figure are shown consecutive diagrams from the same engine when the ignition tube flame was heating the tube too near its top end, causing late ignition; in fact, so late that it began some time after the piston had begun the working stroke. The lower diagrams were taken during the adjustment of the flame, the highest diagrams being obtained after the hot region had been lowered.

Fig. 231 gives diagrams taken at three-quarters full load. When an explosion has been cut out by the governor, the water jacket has had double the time to cool the cylinder, and the two charges of air coming into the cylinder also help to cool the cylinder. When the next explosive charge enters the cylinder, the latter will not be able to heat the charge in the same manner as the hotter cylinder during the previous cycle, and the charge will not so readily ignite. This is emphasised when two or more ignitions are cut out by the governor, causing later ignition and a much lower

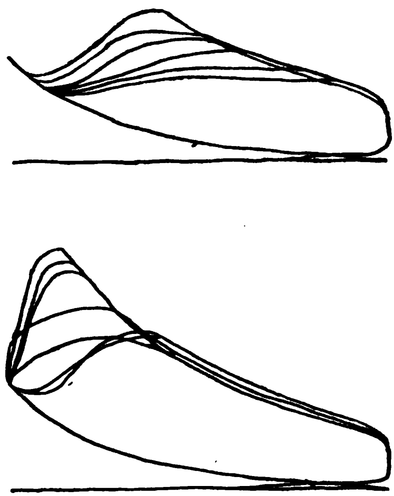


FIG. 230.—Adjustment of Ignition Tube Flame.

explosive pressure. Also, during the cutting out of one or more explosions, the hot products of combustion usually remaining after exhaust, are more or less swept out, thus further assisting to cool the cylinder. The dotted diagram in Fig. 231 represents the state of affairs just described, but it is much better illustrated in Fig. 232. Here the engine is running light, and the explosions are sometimes so far apart that the cooled cylinder does not permit an explosion, but compels the charge to *burn* almost continuously throughout the stroke.

Fig. 233 shows further diagrams illustrating very slow burning. One of the expansion lines never reaches the normal terminal pressure, and here probably the charge continues to burn throughout the whole of the stroke, and

possibly in the exhaust pipe, when the conditions are suitable. If, as is quite possible, a charge of gas is not

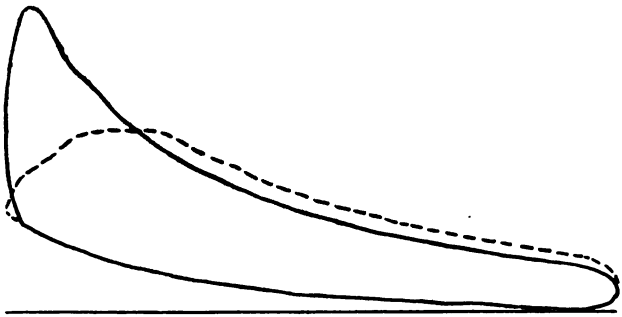


FIG. 231.

ignited, and is turned into the exhaust pipe unexploded, and there follows immediately after it, a very slow burning

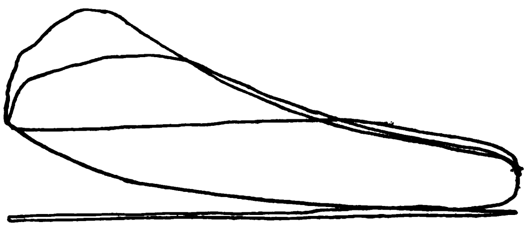


FIG. 232.—Gas Engine Running Light, Slow Burning.

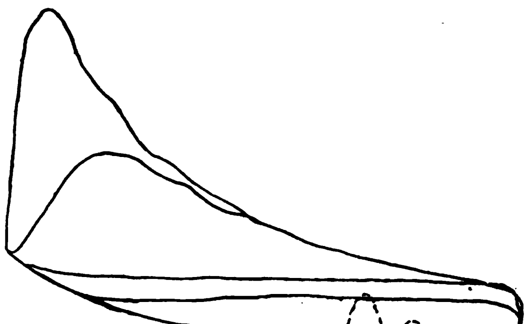


FIG. 233.—Gas Engine, No Load.

charge such as that just described, there is every probability of the virgin charge in the exhaust pipe being

exploded by the succeeding slow burning charge, producing a loud report known as a back fire. This happened in Fig. 233.* Slow burning is also produced by diluting the charge with more air; but the dilution is accidental in most engines, while the cooling of the cylinder is a natural and unavoidable phenomenon.

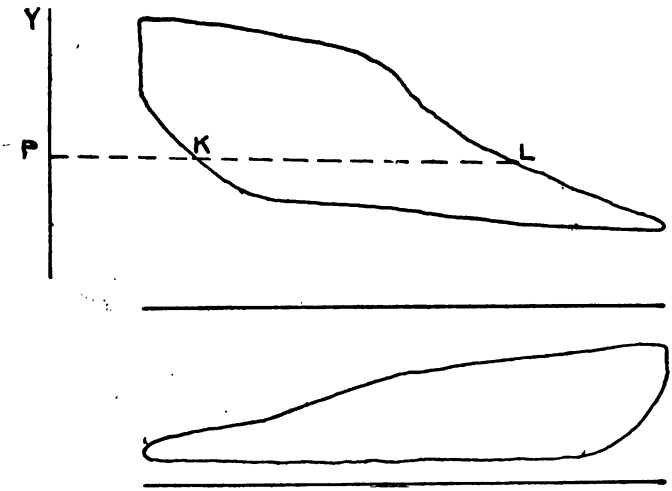


FIG. 234.

Combining Diagrams from Compound Engines.— It is sometimes useful to reconstruct a set of indicator diagrams, so that they appear in the new figure as if the steam had acted all the time on the low-pressure piston only. This is done by plotting the several diagrams upon the *same volume base*, but with their respective pressures represented by ordinates as usual.

Two diagrams, one from each cylinder, are given in Fig. 234 the upper (high-pressure) one was taken with a 100-spring, and the lower (low-pressure) one with a 40-spring. The latter diagram is reproduced in the lower part of the next figure, but it is so situated that the vertical axis O E is to the left of the admission line, by a distance O N, representing the clearance volume of the low-pressure cylinder.

Thus, the length N M represents the volume swept out by the piston, and O N the clearance volume. Hence, at the pressure O R in the low-pressure cylinder, the actual

* The dotted part of Fig. 233 is due to the charge exploding during the suction stroke.

volume of steam which is expanding in the cylinder is represented by R T. Of this amount the volume R Q was imprisoned by the slide valve when compression commenced, and therefore the steam whose volume is represented by Q T entered the cylinder as new steam while the steam port was open, and it will pass out of the engine when the exhaust port is opened.

The high-pressure diagram must be reproduced to the same scale of pressures as the low-pressure diagram in Fig. 235, therefore each pressure ordinate of the high-pressure diagram in Fig. 234 must be multiplied by $\frac{100}{40}$ before it is plotted in Fig. 235. Further, the volume swept out by the piston in the high-pressure cylinder will be a fraction of that swept out by the low-pressure piston for the same

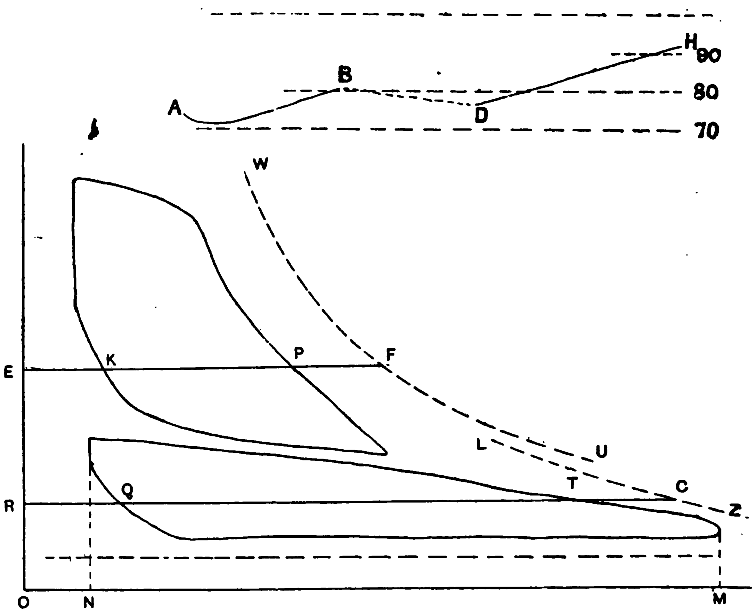


FIG. 235.

percentage of stroke; hence the length of every horizontal line in the high-pressure diagram, Fig. 234, must be multiplied by that fraction before being reproduced in Fig. 235. Thus the cylinder diameters were 20 in. and 30 in. respectively, and for the same percentage of stroke the

respective pistons would sweep out volumes in the ratio of 20^2 to 30^2, or 1 to 2·25. Therefore, the horizontal lengths in the high-pressure diagram will be reduced in the ratio of 2·25 to 1 before reproduction in Fig. 235. For example, the clearance volume in the high-pressure cylinder has been indicated in Fig. 234 by the

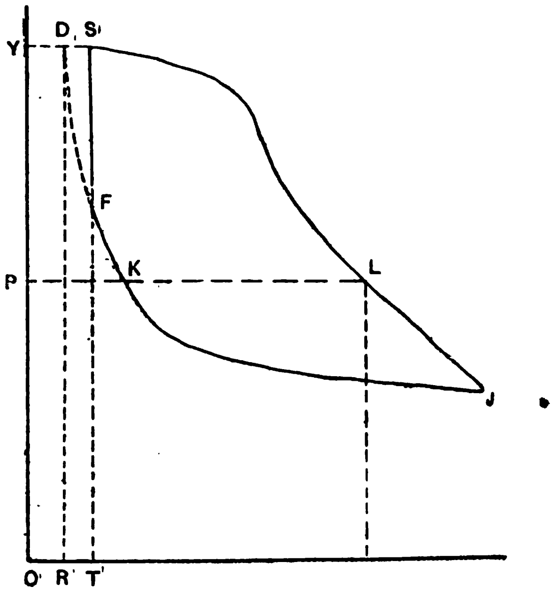


FIG. 236.

position of the axis, P Y. The absolute pressures at K and L have been multiplied by $\frac{100}{40}$ and then set off at O P, Fig. 236.*

Then P K and P L (Fig. 234) have been divided by 2·25, and set off at P K and P L (Fig. 236). In this way the high-pressure diagram is reproduced in Figs. 236 and 235, showing the work done in the high-pressure cylinder to the same scale as the low-pressure diagram.

Losses shown Graphically on Combined Diagram.—Referring to Fig. 236, it will be seen that if the imprisoned steam had been compressed during the return stroke up to the initial pressure, the compression curve would be

* The high-pressure diagram of Fig. 235 has been first reproduced in Fig. 236 for the sake of clearness before transferring to Fig. 235.

N

K F D; and at the beginning of the stroke, the clearance volume would be quite full of steam at the admission pressure. Any steam which then came into the cylinder would push the piston down the cylinder and do useful work, the cushion steam merely acting as an elastic buffer, giving out as much work during expansion as was done *upon it* during compression, and consequently not contributing to the net useful work done upon the piston.

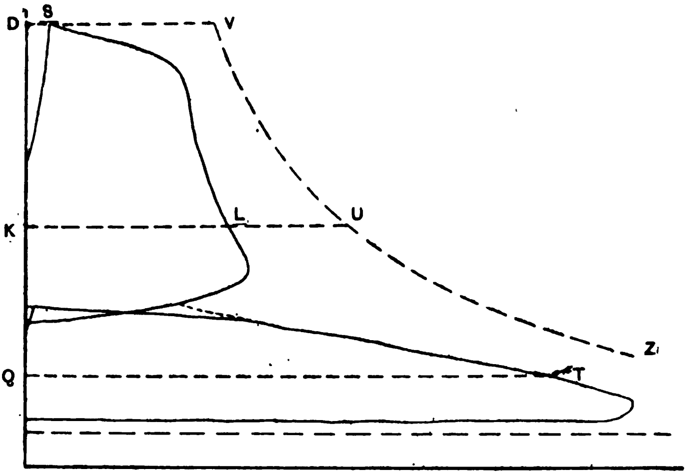


FIG. 237.

When the cushion steam is not compressed by the piston to the boiler pressure, the *incoming steam* completes the compression, just in the same manner as the piston did in the previous paragraph.

At the beginning of the forward stroke, the volume of steam in the cylinder is represented by Y S, and the volume of *cushion steam* at the same pressure is represented by Y D; therefore, S D represents the volume of steam which has come into the cylinder to completely fill up the clearance volume at the initial pressure, and the triangular area F D S represents the amount of work which has been lost in so doing.

Now plot a new combined diagram leaving out of account the imprisoned cushion or buffer steam. In Fig. 237 the letters for the several points correspond with those in Fig. 236. The diagram areas represent the useful work done on the piston, the areas between the vertical axis and the

diagram areas represent the waste due to filling up the clearance volumes with new steam, and the area between the diagram areas represents the loss between the cylinders, due to throttling, radiation, condensation, etc.

If the steam consumption or feed water has been measured, set off D V to represent the volume of the steam per stroke entering the engine at boiler pressure, and through V draw an adiabatic line V U Z. The area between

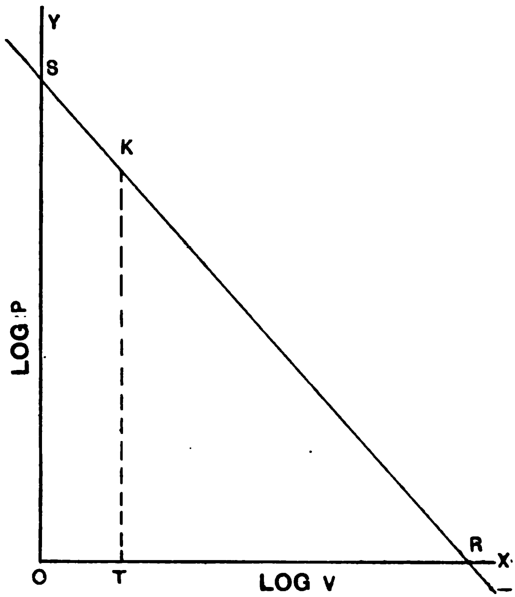


FIG. 238.

this line and the diagram areas represents the loss due to condensation in the cylinders *over and above what would have taken place with adiabatic expansion in a non-conducting cylinder.*

The adiabatic line may be drawn with the aid of a temperature-entropy diagram or by means of the equation,

$$P V^{\frac{10}{9}} = \text{constant.*}$$

This curve is most easily plotted in the following

* This equation, due to Rankine, has been shown by Zeuner to be more accurate when the exponent $\frac{10}{9}$ is replaced by $1{\cdot}035 + {\cdot}1\,x$ when x is the dryness fraction of the steam at the beginning of expansion.

manner. Taking logarithms of both sides of the equation, we have—

$$\log P + \frac{10}{9} \log V = \log \text{constant}.$$

This equation represents a straight line* if the logarithms of the pressures are plotted as ordinates upon a base representing the logarithms of the corresponding volumes.

Only two points are required in the above straight line. The constant is obtained from the pressure and volume of the steam at the point V, Fig. 237. Now put P = 1 lb. per square inch, then log P is zero, and

$$\log V = \frac{9}{10} \log \text{constant}.$$

Where log P = 0, that is along O X (Fig. 238), set off O R to represent ·9 (log constant); then R is a point in the desired line.

Next, when V = 1, log V = 0 and log P = log constant.

Log V = *o* along the line O Y, hence set off O S to represent (log constant) and join S R.

To obtain points on the adiabatic curve V Z, Fig. 237, take any point T in the base line, Fig. 238, and measure off the vertical intercept T K. Find the numbers whose logarithms are represented by O T and T K, and these numbers are the corresponding volume and pressure. Plot these in Fig. 237, and repeat the operation until sufficient points are obtained.

The Quality of the Steam during its passage through the engine may be obtained from the combined diagram, Fig. 235. It is first necessary to construct a combined diagram similar to Fig. 235, which is the mean of a series of pairs (crank and head end) taken during a trial. Then from the steam consumption determine the weight of steam used per stroke.

Draw a horizontal line such as E K P F through the compression and expansion curves. E K represents the volume of cushion steam, K P the volume of actual *steam* passing through the engine per stroke; and if W F U is a saturation curve, P F must represent the volume of steam which has been condensed and exists as moisture at that

* See the author's Tables and Data for Engineering Laboratories, p. 60.

pressure. In other words, P F represents the moisture in the steam, or wetness, and E P the dry saturated steam in the cylinder and clearance space; hence the fraction of the *whole*, which is dry saturated steam, is

$$\frac{EP}{EF}.$$

This is called the dryness fraction. From K set off horizontally K F to represent the volume of steam per stroke as obtained from the steam consumption. The same total weight of the mixture of steam and water will exist throughout the stroke, hence it is only necessary to find the volume of that weight of dry saturated steam at the various pressures and to plot them to the right of the compression curve. Should the compression curve not exist throughout the whole range required, a rectangular hyperbola would be drawn in either direction so as to form a continuation of the compression curve. This has been done in Fig. 236.

The same process must be repeated for the low-pressure cylinder, and as the ratio of the clearance volume to that swept out by the piston is generally different for the two cylinders, a different saturation curve will have to be used, but found in precisely the same manner as described above for the high-pressure diagram. L C Z is the saturation curve for the low-pressure diagram.

The quality or dryness curve is shown at A B D H, Fig. 235, at the top of the diagram, the numbers at the right indicating percentages of dryness. These curves do not by any means always present the same appearance, and sometimes trend downwards instead of upwards.

The Hyperbolic Curve.—It is found that the *expansion curves* of the combined diagrams, when taken from a well-designed engine of not too small a size, and in good order, very nearly coincide with a rectangular hyperbola, and hence the curve is often used to roughly determine how the engine is working. Examples of its use have already been given in connection with the analysis of the diagram, but one more is here given in Fig. 239, showing throttling in low-pressure ports, and perhaps leakage past the piston.

The curve is most easily applied to the diagram by drawing a number of hyperbolæ on a piece of tracing cloth, and placing the tracing cloth over the combined

diagram, the axes of the curve coinciding with the lines of zero volume and zero pressure. A number of these hyperboles have been drawn in Fig. 240, and can be traced on cloth or tracing paper, which may then be superposed on the indicator diagram.

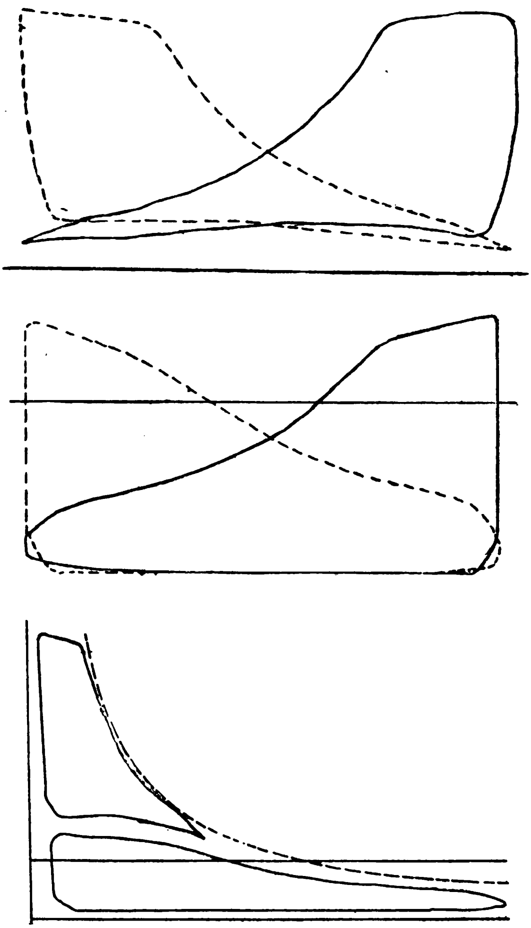


FIG. 239.—Diagrams from a Side-by-side Compound Engine, with Corliss Valves. Revolutions 33.

A set of diagrams having rather a peculiar appearance when combined is shown in Fig. 241. They were taken from one of Stewart and Nicholson's continuous-expansion

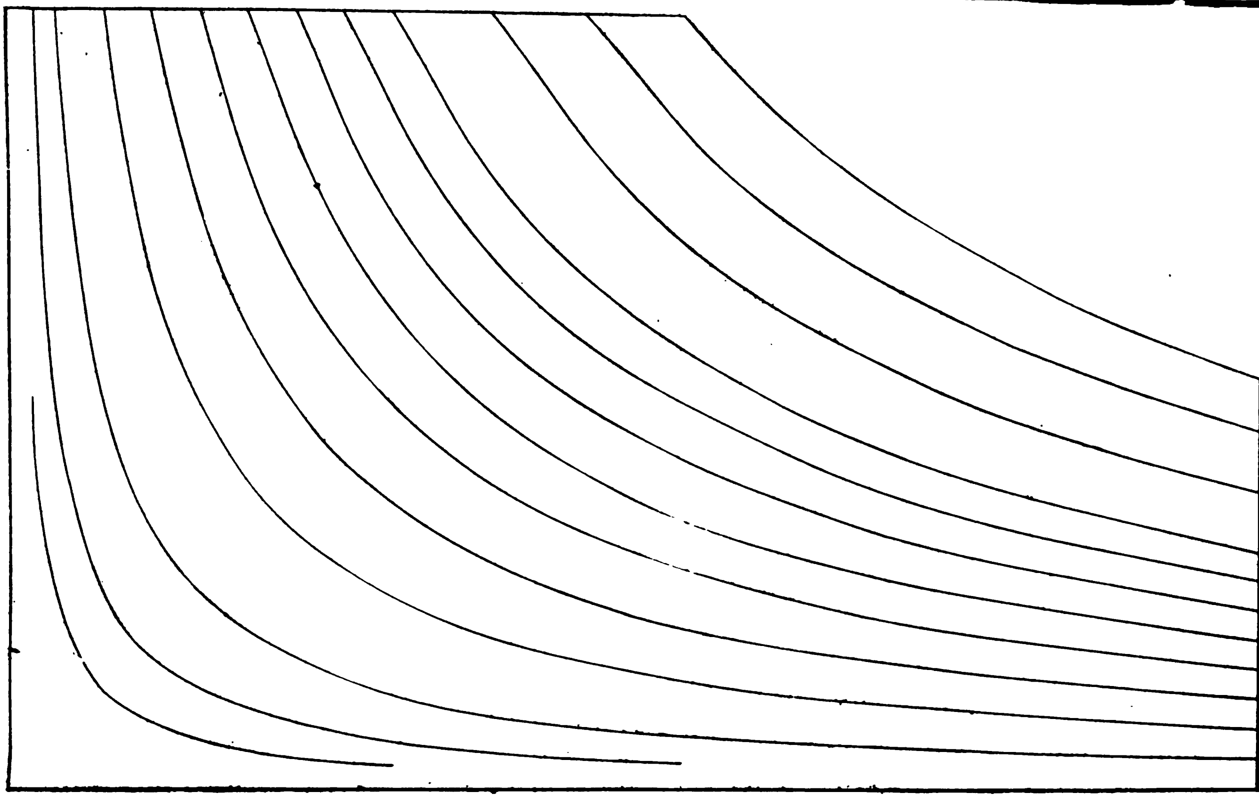


FIG. 240 —Rectangular Hyperbolæ for use with an Indicator Dirgram.

engines, in which there is a port half-way down the high-pressure cylinder, so that after half stroke the steam expands simultaneously in both cylinders.

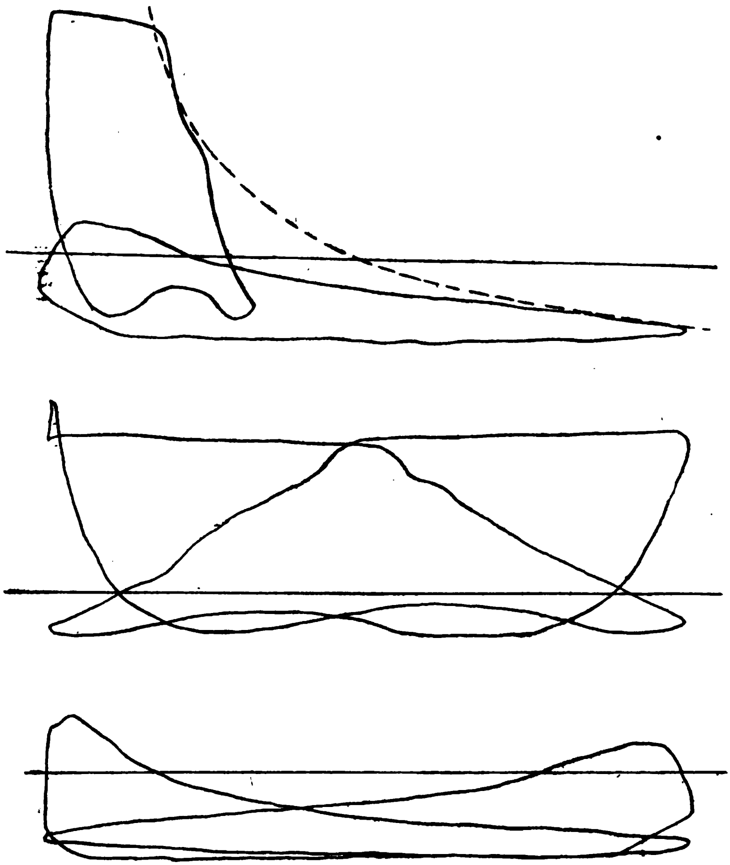


FIG. 241.—Diagrams from a Marine Engine on Stewart and Nicholson's Patent Continuous Expansion System.

Clearance.—The clearance volume is best found by filling up the cylinder end with water, or calculating it from the working diawings. It may be found approximately from the indicator diagram if there is no leakage as follows:—The line of zero pressure E P M, Fig. 243, is known from the atmospheric line. Draw any line,

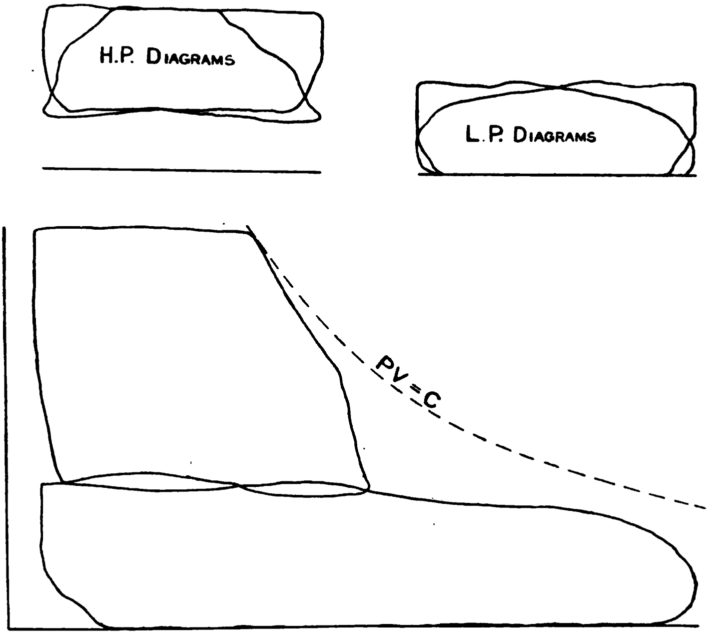


Fig. 242.—Diagrams from a Worsdell Compound Locomotive.

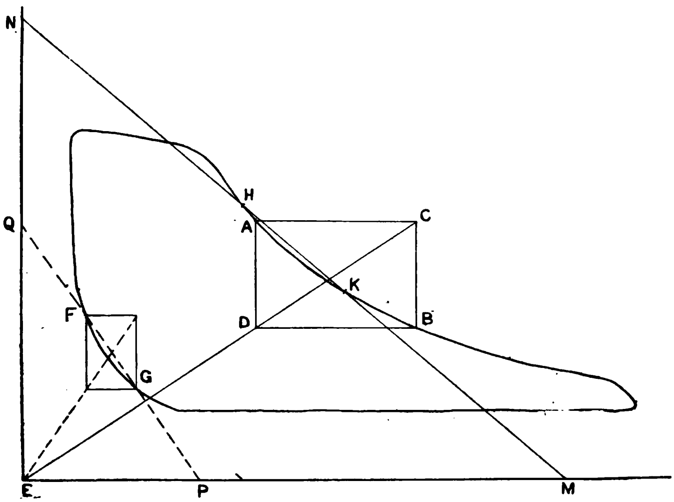


Fig. 243.—Clearance Volume.

N H K M, cutting the expansion or compression curve, then N H = K M. Through N draw a vertical cutting E P M in E. Or draw a rectangle with opposite corners A and B on the expansion or compression line. Draw the opposite diagonal C D and it will pass through E.

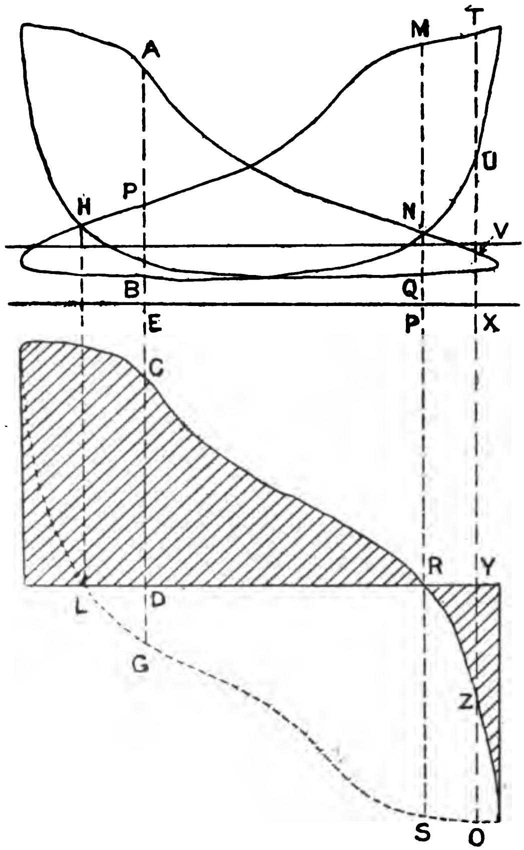


FIG. 244.

Resultant Pressure on Piston.—At any point in the stroke of an engine, such as E (Fig. 244), the forward pressure is represented by E A, and the pressure on the other side of the piston by E B, therefore the effective forward pressure is E A − E B = B A; the line E P X

being the line of zero pressure. On the line L D R Y plot

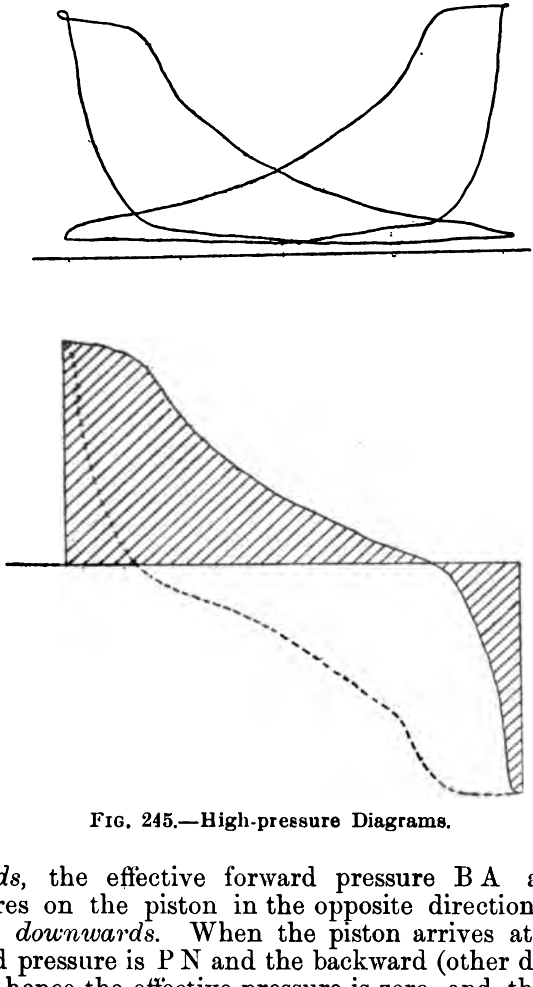


FIG. 245.—High-pressure Diagrams.

upwards, the effective forward pressure B A at D C. Pressures on the piston in the opposite direction will be plotted *downwards*. When the piston arrives at P, the forward pressure is P N and the backward (other diagram) is P N, hence the effective pressure is zero, and, therefore, the outline of the effective pressure curve crosses the base line at R.

With the piston at X, and moving in the same direction, the forward pressure is V X, and the backward pressure is U X, the effective or resultant pressure being U V in the backward direction, and consequently must be plotted downwards at Y Z.

The shaded area above the base line represents positive work done by the steam on the piston, and the shaded area below the base represents negative work. The difference of the two should be equal to that of the indicator diagram.

If the same process be carried out for the return stroke the dotted outline will be obtained in the lower part of the figure; and the net work done during the two strokes is represented by the area included between the two curves so found.

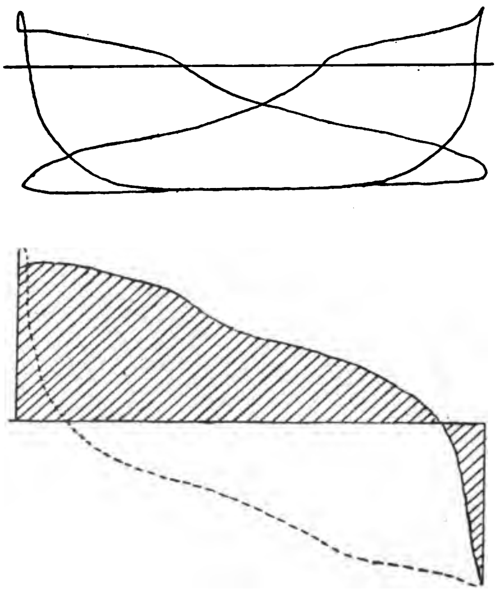


FIG. 246.—Low-pressure Diagrams.

In Figs. 245 and 246 are shown the high and low pressure diagrams of a Corliss engine redrawn in the manner just explained.

The obtaining of the resultant pressure is useful in the design of flywheels and the drawing of crank effort diagrams.

CHAPTER XIV.

DIAGRAM AVERAGING AND MECHANICAL EFFICIENCY.

Mean Pressure from Diagram.—The mean height of a given area may be obtained (1) by splitting the figure up into a series of strips of equal width parallel to the direction in which the mean height is measured, and taking the mean height of each strip, and finally the average of these means; (2) by obtaining the area with a planimeter, and dividing the area by the length of the figure, measured in a direction perpendicular to that of the mean height; or (3) by a planimeter so arranged that it gives the mean height direct.

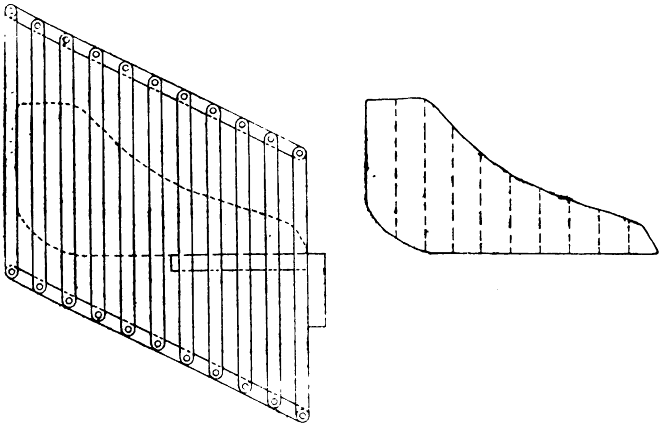


FIG. 247.— Parallel Rule for Dividing-up Diagrams.

Of these methods the last is that which is usually adopted, on account of the speed and accuracy with which the mean height can be found. The splitting up of the figure can be more easily and quickly done by means of a special parallel rule, which is sometimes supplied with indicators. It is shown in Fig. 247, placed over a diagram in position ready to draw the parallel lines, by which the diagram is divided into ten parallel strips of equal width.

Let the mean height of a strip be denoted by h, with a subscript indicating the number of the strip counting from the left. Also let l represent the length of a vertical intercept, with a subscript indicating its position from the left end of the diagram.

The first strip is bounded vertically by l_1 and l_2. Similarly the second strip is bounded by l_2 and l_3, and the last by l_{10} and l_{11}. The mean height of the diagram is approximately the average of the mean heights of the strips; and

$$= \frac{h_1 + h_2 + h_3 + \dots\dots\dots + h_{10}}{10}$$

But

$$h_1 = \frac{l_1 + l_2}{2}$$

and

$$h_2 = \frac{l_2 + l_3}{2}$$

also

$$h_{10} = \frac{l_{10} + l_{11}}{2}$$

Substituting these values for h, &c., we have the mean height of the diagram

$$= \frac{l_1 + 2l_2 + 2l_3 + \qquad + 2l_{10} + l_{11}}{20.} \dots\dots\dots\dots\dots$$

or after dividing numerator and denominator by 2, we get the mean height

$$= \frac{\tfrac{1}{2}\text{ (first ordinate and last ordinate)} + \text{all other ordinates}}{\text{Number of strips into which the figure is divided.}}$$

The method just described is based upon the assumption that each of the strips is a trapezoid. The area of a trapezoid equals the product of its width and its average height. The latter can be read off direct by measuring the height at the centre of a strip. It is then only necessary to draw the vertical lines through the centres of the strips, without actually drawing the dividing lines between the strips.

Looking at Fig. 247, it will be noticed that if the middle heights were drawn in, the first one would be half the width of a strip away from the left side of the diagram. The next one would be the width of a strip from the first, and so on, until at the other end, the last height would be half the width of a strip from that end. In Fig. 248, the

line A N has been divided at the points B, C, D, . . . K and L, in the manner stated above, such that those are the middle points of the ten equal parts into which A N is supposed to be divided.

Now, different diagrams will be of different lengths, hence it will be necessary to make a figure which will give the same kind of division whatever the length of the indicator diagram. Take any point in A R produced, and

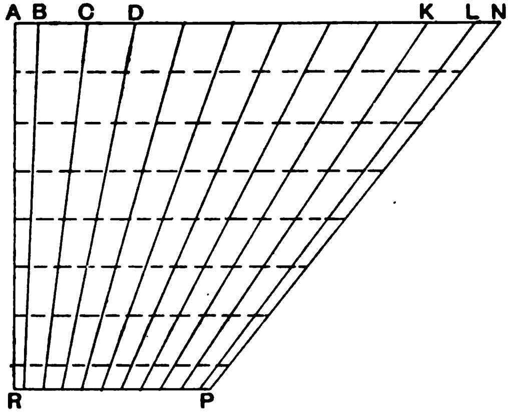


FIG. 248.

join it to A, B, C, etc. We then have the inclined lines shown in the figure.

Now take the indicator diagram and draw two lines at right angles to the atmospheric line, and touching the diagram at each end, such as E A and T V Fig. 249. Place the indicator card upon the radial diagram such that the atmospheric line is parallel to one of the dotted horizontal lines, the line T V coinciding with S R and the point A in the line E A on the right hand radial line N P. Pin down the indicator card, and draw across the indicator diagram a series of vertical lines from the points where the radial lines cut the line V A. These are shown dotted in the figure, such as B C D. Measure off the intercepts such as C D, and take their average by adding them together and dividing their sum by their number. This is the mean height of the diagram. If these heights are measured off with indicator scales, the average so found will be the actual mean effective pressure on the piston in pounds per square inch.

The Coffin Averager and Planimeter.—Generally a number of these diagrams have to be measured up at one time, and then it is expedient to use methods 2 or 3 mentioned above. The Coffin averaging planimeter, shown in Fig. 250, is an instrument by which the mean height may be found direct. It is sold in this country by Hartley and Co., of Manchester. It consists principally of a rod, in one end of which is a tracing point E, which is moved over the outline of the diagram. The other end of the rod contains a pin, which slides in the groove in the plate I, on the left of the figure, the pin being maintained in contact with the groove by the weight W. The rod also contains the bearings for the spindle of the graduated

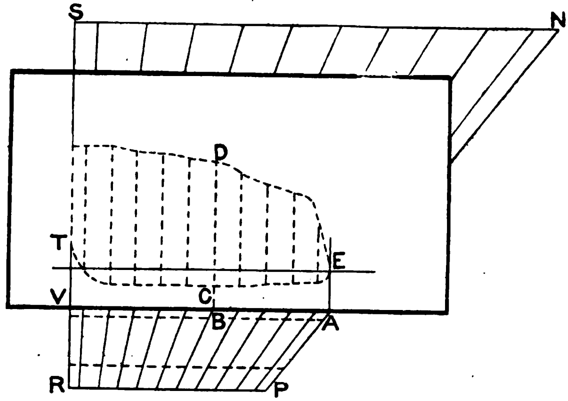


FIG. 249.

wheel, near the lower edge of the figure. The wheel has a small flange at one edge of the rim, which rests upon the strip of paper indicated by the light vertical strip in the figure. The rod is thus supported on three points, namely, the tracing point E, the pin W, and the flange of the graduated wheel. Attached to the board are a pair of clips B and K, of which the latter is capable of being moved parallel to itself by means of the slide at its lower extremity. One of horns of the rod which supports the graduated wheel is divided so as to form a vernier for the more accurate reading of the graduations.

To obtain the mean height of an area (in the figure an indicator diagram) slide the paper containing the area under the two clips B and K, and arrange it so that a

horizontal line in the diagram is parallel to the horizontal edge of the clip B and the left-hand end of the diagram touches the vertical limb of the same clip. In an indicator diagram the atmospheric line is horizontal. Now push the clip K towards the left until its inner edge touches the right-hand end of the diagram (in the figure at E.) Place

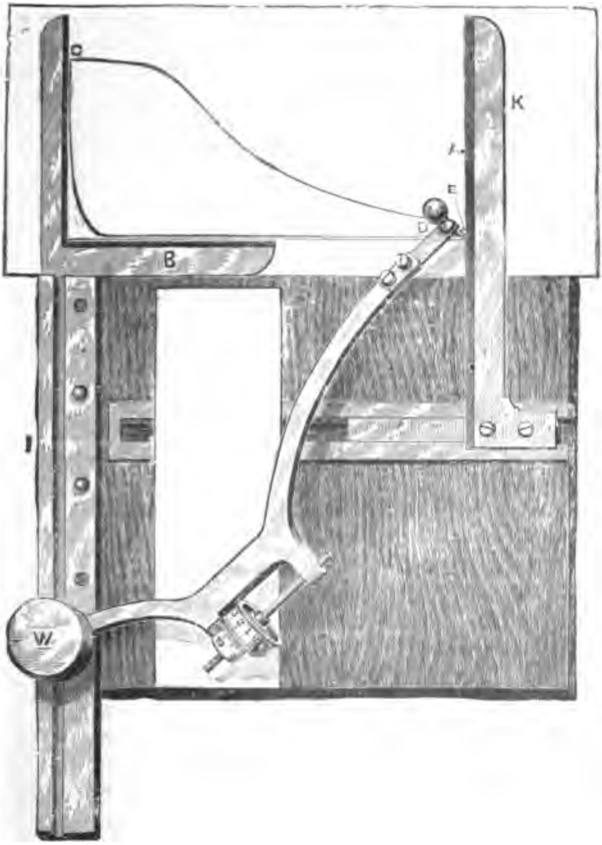


FIG. 250.—Coffin Averaging Planimeter.

the planimeter in the position shown in the figure with the tracing point at E where the clip K touches the diagram. Press the head D of the tracing point so as to make a mark at E, and then raise one of the horns while the graduated wheel is turned to zero. Care should be taken in this operation that the fingers are free from moisture so that

the steel wheel will not become rusted. It would be much more convenient if the rim, with the exception of the flange, were made of white celluloid, as in the Amsler planimeter, then the rusting would be avoided and the reading of the graduations rendered comparatively easy. The outline of the diagram is now *carefully* traced over with the tracing point in clockwise direction until the starting point E is again reached. Now slide the tracing point along the edge of the clip K (keeping the eyes on the rolling wheel) until the wheel indicates zero. The tracing point is now at A. Then E A is the mean height of the diagram. The tracing is most easily accomplished when the operator is comfortably seated, with his arms resting on the table which support the instrument. Diffused light is also much pleasanter to work in than light emanating from a single source (such as a gas flame), on account of the dark shadows cast by the latter, which, in some positions, cover up the faint lines of the indicator diagram. If a single source of light is inevitable, the position of the diagram should be so arranged that the shadow cast by the tracing point will not lie along an appreciable length of the outline.

Some people prefer not to set the rolling wheel to zero at starting, but to note the reading; and after tracing the figure to slide the tracing point along the edge of the clip K until the wheel indicates the same reading again; at which time it will have arrived at A. This latter method is slightly more accurate than the former, and the wheel does not become rusty, due to the moisture of the hands.

The Amsler Planimeter.—If the end W of the rod in the Coffin averager be made to travel over the arc of a circle instead of along a straight line, we have at once the Amsler planimeter. This is accomplished by attaching the end W to a radius rod, and it can be shown* that the rolling wheel records the area of the figure whatever be the path of the end W of the rod. Fig. 251 shows a plan and elevation of the ordinary type of the Amsler instrument, for which the author is indebted to Messrs. Schäffer and Budenberg. The free end of the radius rod contains a needle point C, which acts as the centre round which it turns. It is loaded with the weight G, to maintain it in position during use. It is preferable to use the instrument on a drawing board previously

* See "Treatise on Engine and Boiler Testing," by the Author.

covered with a *smooth* hard drawing paper or cardboard. Rough paper should not be used. The other end of the radius rod is pivoted to a saddle, which slides along the

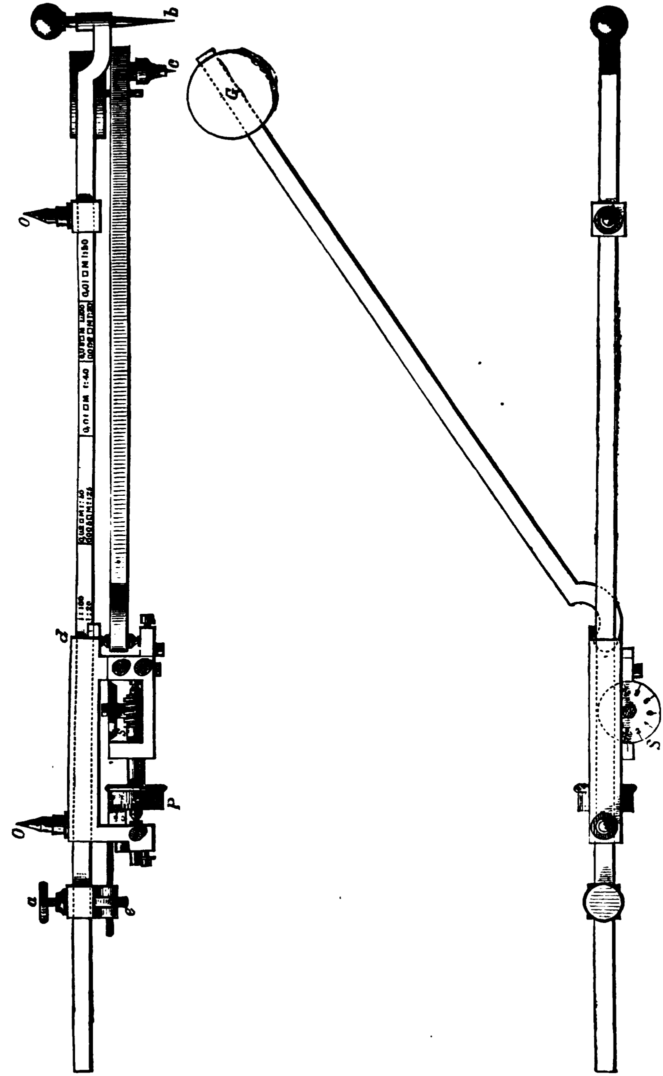


FIG. 251.—The Amsler Planimeter.

square rod shown in the horizontal position in plan, and which carries the tracing point *b*. This saddle also carries

the recording wheel P, with its spindle and worm, together with the recording disc S, and a worm wheel (not shown in the figure). The saddle, which is secured to the tracing rod by means of the clamping screw *a*, contains the fine adjustment screw *e*, by which it can be accurately placed in any position along the tracing bar. There are two points O O, one on the tracing bar and the other on the saddle. These are brought into use when it is required to obtain the mean height direct. The clamping screw *a* is screwed back, and the saddle pushed along the tracing rod until the distance between the points O O equals the length of the figure or diagram; the fine adjustment being accomplished by the milled head *e*, after the clamping screw

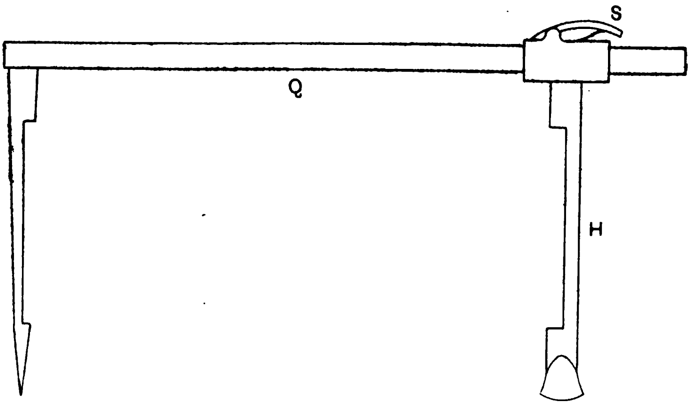


FIG. 252.—The Goodman Averager.

a has been tightened up again. Now, place the instrument on the drawing board, and with the tracing point *b* make a mark or indent, somewhere on the outline of the figure or diagram. Read the disc S, the graduated wheel P, and then the vernier. As the wheel is divided into 100 equal parts, the main divisions will be read for the second figure, and the sub-divisions, for the third; while the vernier will give the fourth. Following this order, assume, for example, that the reading is 212·5, the decimal being placed before the vernier figure. Now, trace the outline of the figure until the tracing point reaches the indent again. The reading is now, say, 301·7. The difference of these readings is 301·7 − 212·5 = 89·2, and, dividing this difference by 40 (the constant of this instrument), we obtain 2·23 in. as the mean height of the diagram. One great advantage of this

instrument is that the actual mean height is given direct from the wheel readings without the use of any scale, and to a degree of accuracy which a scale could not indicate. It is advisable to start the tracing point a little distance from the indent on the outline, and trace the outline up to the indent before taking the first reading. In this way a slight error due to backlash can be avoided.

The Goodman Averager.—Another form of instrument for obtaining the mean height of a diagram is Professor Goodman's modification of the hatchet planimeter. It is shown in Fig. 252, and consists of a bar Q, with an adjustable limb H, the extremity of which is shaped like a hatchet held tight upon the bar by the spring S, and another leg rigid with the horizontal bar and used as a tracing point. In using the instrument, the movable leg is shifted along the bar until the two legs just encompass the length of the diagram, as shown in Fig. 253.

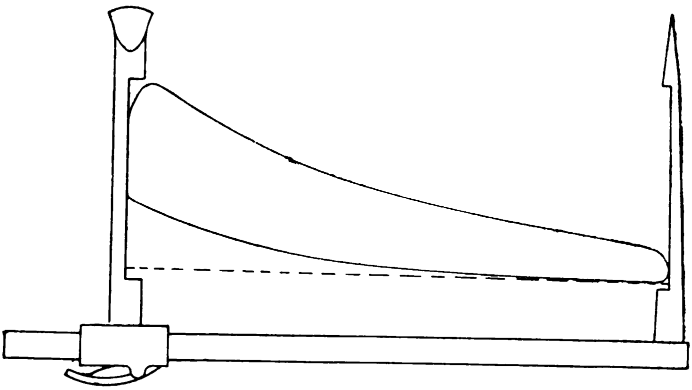


FIG. 253.—Sketch showing Application of Goodman Averager.

Now, draw a line from the point A (the centre of gravity of the area) to the outline as in Fig. 254. Place the bar of the instrument approximately at right angles to the line from A—that is, along the dotted line—and the tracing point at A. Press the hatchet end, making an indent at B, and then trace the outline once, beginning at A and returning to it again, preferably always in the same direction for all diagrams such as that shown by the arrows in the figure. When the tracing point has arrived at A,

after going once round the figure, press the hatchet again, making the indent C.

Now, unpin the diagram, all the while keeping the tracing point at A and the hatchet at C, and turn the diagram round A, through approximately two right angles into the position shown in Fig. 255, and then retrace the outline of the diagram in the *opposite direction* (as shown by the arrows), until A is reached again, pressing the hatchet at the end of the operation, making the indent D. The point A was selected by the computer's judgment, and is necessarily not often exactly at the centre of gravity of the area. This and other small errors cause the slight displace-

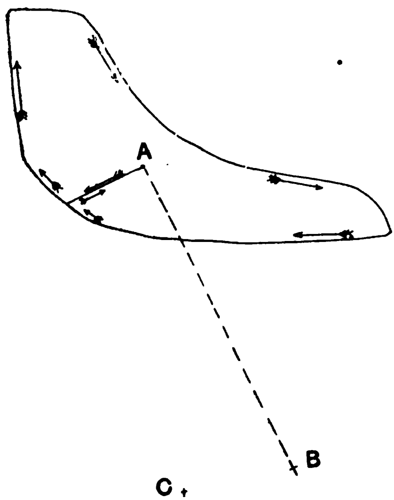


FIG. 254.

ment of the finishing indent D from the initial indent B. A point E is selected half way between their centres, and the distance C E measured with a scale. This distance is the mean height of the diagram, and, if measured with the scale of the spring used, will give the mean pressure direct.

It is necessary when using this instrument to keep its plane vertical, and to take great care of the hatchet end. The operation should be carried out on a horizontal plane, preferably a drawing board covered with good *smooth* drawing paper. It is preferable to draw the line from A to the boundary of the diagram, either parallel to the

atmospheric line or in such a way that it is in the direction of the longest way of the diagram. The instrument is made by Messrs. Jackson Brothers, of Leeds.

Hand Speed Counters.—The Tabor pocket revolution counter is shown in Fig. 256. The diamond point centre on the left of the figure is inserted into the centre of the shaft whose speed is required, and the friction between the male and female centres compels the small horizontal spindle to rotate at the same speed as the shaft, and the worm on it turns the graduated disc through one tooth or division for every revolution of the spindle. The number of revolutions in a given time is obtained by reading the

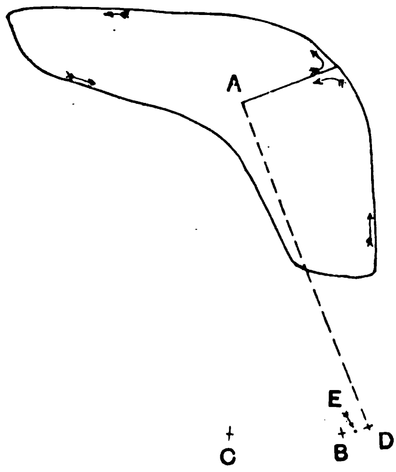


FIG. 255.

units and tens opposite the hand, the hundreds and thousands being indicated through the aperture in the disc. In the illustration the counter reads 4535. The under dial reads only up to 49, so that the instrument will count up to 5,000. To set the instrument to zero, draw out the spindle until the dials can be rotated by the fingers; then set them to zero.

The counter shown in Fig. 257, is designed to record revolutions and reciprocations. A number of figured discs are threaded upon a spindle, and each disc, except the first, gears with, and is driven by, a small pinion loose on a lay spindle. Each disc also gears with the next

pinion in such a manner that, after turning round once, it moves its next neighbour by means of the pinion through the interval between two consecutive numbers. The pinions and discs are so shaped that they lock one another

FIG. 256.—Tabor Pocket Revolution Counter.

in position, and only permit of the requisite motion to enable the instrument to count the revolutions or reciprocations, as the case may be. For the latter, the rod B is used to move the counter wheels while the dotted position perpendicular to B shows its position for counting revolu-

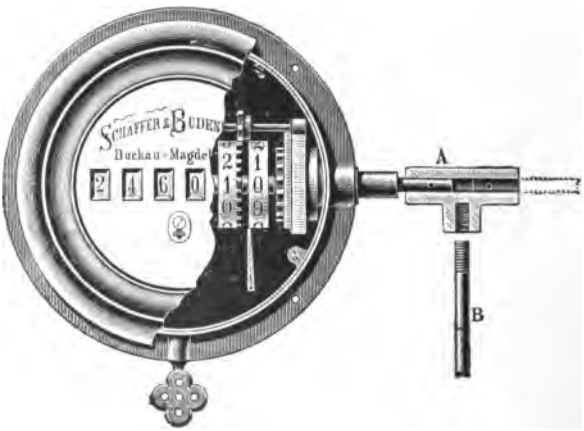


FIG. 257.—Schaffer and Budenberg's Counter.

tions. Messrs. Schäffer and Budenberg also make a small pocket counter of the same type, which is used in a manner similar to that in Fig 257.

Mechanical Efficiency.—As there are always some internal resistances in a machine under all actual con-

ditions of working, there will be a certain portion of the energy received by a machine which will be used in driving the machine against these internal resistances, and therefore only a fraction of the energy received by a machine will be given out by the machine in doing useful work.

The ratio:

$$\frac{\text{Useful work done by a machine}}{\text{Total energy received by a machine}}$$

is called the *mechanical efficiency* of the machine. If we take the work done during one minute then we may express the mechanical efficiency as:

$$\frac{\text{The horse power at which a machine gives out energy}}{\text{The horse power at which a machine receives energy}}$$

It is the numerator of this fraction that we are now concerned with, and it is measured by a *dynamometer* in conjunction with a *speed indicator*. Dynamometers are of two kinds—friction brakes, sometimes called absorption dynamometers, and transmission dynamometers. In the former class, the energy given out by a machine is used to work against some frictional resistance, in which case the energy is converted into heat, and, in the second class, the energy is transmitted through some form of mechanism which registers the force exerted by some part of the machine in doing useful work.

The Rope Friction Brake is the most convenient form of absorption dynamometer under ordinary circumstances, and is by far the cheapest and easiest to construct. A simple form, as applied to a small engine, is shown in Fig. 258, where one end of the brake rope supports a load W, and the other end X is attached to the spring balance S, which is anchored to the floor or other fixture. In this case the rope is in contact with the wheel through half the latter's circumference only, and is prevented from slipping off the wheel laterally by two or more guide blocks Z. These blocks are preferably made of some hard wood (the author has used teak), and about a quarter or an eighth of an inch longer than the width of the wheel rim. Small brass

plates overlapping the rim are attached to the ends of these blocks by screws to prevent lateral movement. If these should make an unpleasant grating noise by the rim rubbing continually against them, small pieces of stout leather may be substituted for them with advantage. Grooves are cut in the under side of each block to receive the rope, but these should not be quite so deep as the rope is thick, or the block itself will rub on the rim of the wheel, producing a noise similar to that previously spoken of. Each part of the rope is fixed to the block by fine wire, care being taken that the wire itself does not touch the rim of the wheel, but is threaded between the strands of the rope. The rope should be as small as possible con-

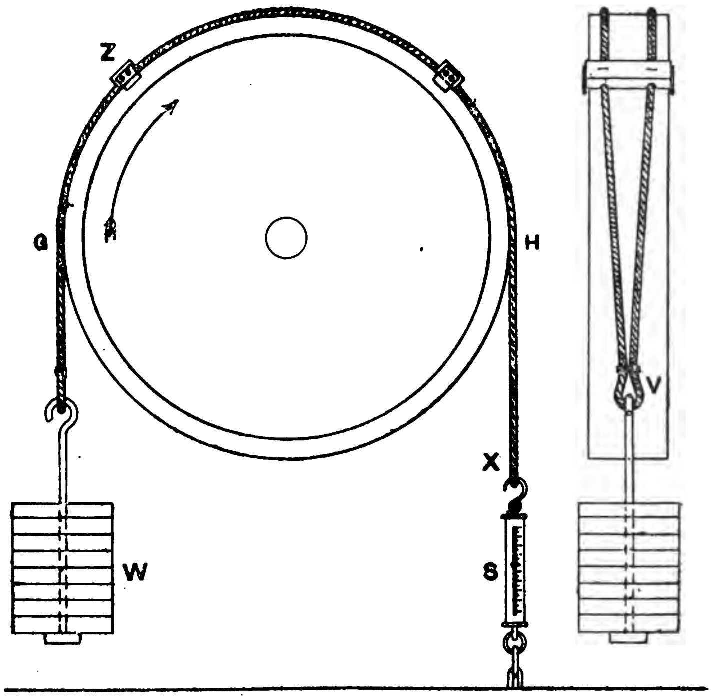


FIG. 258.—Rope Brake for Small Powers.

sistent with strength, as flexibility may be sacrificed with a rope of much larger diameter. The more flexible the rope, the more easily will it run without oscillation.

It is necessary to bind the rope at one end with copper wire, so as to form a permanent eye, into which the end of

the weight pillar is hooked, while the other end may be spliced, or only bound with wire to make a joint; but it is not advisable to further bind it as shown at V, as, should one part of the rope stretch under the load more than its companion, the guide blocks Z tend to twist about a radius of the wheel as axis, and the brass clips tend to grip the wheel, resulting in much vertical oscillation of the weight W. The rope can often be run in the dry condition without trouble, but should this produce unsteadiness, it may be smeared with a little Russian tallow.

The tension of the spring balance acts in the direction of motion, while the load on the other end of the rope acts against the motion of the wheel. Hence the net resistance of friction is

$$(W - S) \text{ lb.}$$

When W represents the pull on the load end of the rope at G, including the weight of the load pillar and that part of the rope from G downward, and S represents the pull on the other end at H which includes the weight of the rope from H down to X. As the spring balance is turned upside down, as shown in the figure, its weight will be included in the reading of the scale opposite the pointer, so that there will be no necessity to make a correction by adding on its weight. Generally the piece of rope below G on one side balances the piece below H on the other, so that the ends may be left out of the question; in any case, where the rope is of small diameter, as it should be, the weight of the ends of the rope may be negected in calculations.

The pull on either end of the rope at G and H, acts along the centre line of the rope, so that the effective radius of the brake is—

$$\text{Radius of face of wheel} + \text{radius of rope},$$

and the effective diameter of the brake is twice the effective radius, or—

$$\text{Diameter of wheel} + \text{diameter of rope} = D, \text{ say.}$$

Then the work done per minute by the wheel in foot-pounds in working against friction equals

$$(W - S) \times \pi D \times \text{number of revolutions per minute},$$

when D is measured in *feet*.

The rate at which the wheel gives out energy, or does work, measured in horse power

$$= \frac{(W - S)\,\pi\,D\,N}{33{,}000}$$

when N is the number of revolutions per minute.

Because this is measured with a brake, it is generally called the "brake horse power" of the engine, and represented by B.H.P. Then we may write

$$\text{B.H.P.} = \frac{(W - S)\,\pi\,D\,N}{33{,}000} \quad . \quad . \quad .$$

LOG OF TEST FOR BRAKE HORSE POWER WITH SIMPLE ROPE BRAKE.

Date .... Observer ....
Kind of motor...
Approximate angle of contact in turns ....
Diameter of brake wheel......ft. Diameter of rope......in.
Effective diameter of brake ....ft.
Zero reading of spring balance.........lb. Error.........lb.
Total weight of discs on end of brake rope ....lb.
Weight of load pillar, etc.......................................lb.
Total load on end of brake rope ....lb.
Brake horse power = c N R c = ...

Time.	Revolutions per minute. N.	Spring Balance Reading S lbs.	Effective Resistance of friction. R lbs.	B.H.P. = C.N.R.	Counter Reading.

CHAPTER XV.

Slide Rules and Calculators.—The great saving of time and labour in making calculations that can be effected by the use of the slide rule is well recognised by those familiar with its working. The ordinary straight slide rule, however, is inconvenient to carry constantly in the pocket, and a modified form of this instrument (Fig. 259), capable of being carried at the end of a chain like an ordinary watch, is, in the author's opinion, the handiest for general work. This particular form of instrument is supplied by the Scientific Publishing Company, 53, New Bailey Street, Manchester.

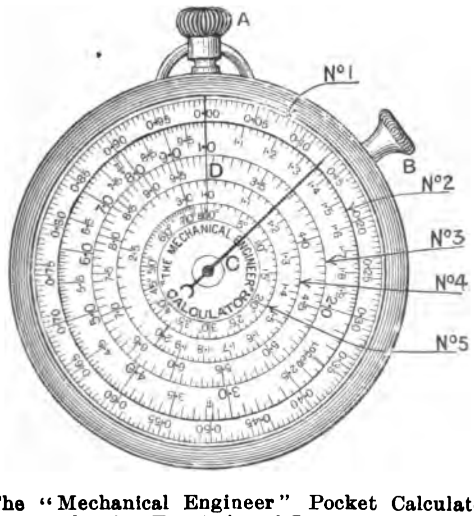


FIG. 259.—The "Mechanical Engineer" Pocket Calculator.—Sketch showing Exact size of Instrument.

Description of the Instrument.—This instrument consists of a revolving dial operated by the nut A, and having on its face five graduated circles or scales, marked in the illustration, for convenience of reference, Nos. 1, 2, 3, 4, 5. The instrument also carries a movable finger C, operated

by the nut B. In addition to these is a fixed radial pointer D. The dial and the movable pointer C, it should be observed, move quite independently of each other.

Description of Scales.—

No. 1 is a scale of logarithms.

No. 2 is the calculating scale for multiplication and division.

No. 3 and 4 is a scale of square roots.

No. 5 is a scale of sines of angles.

Logarithms.—The readings on scale No. 1 are the common logarithms of the numbers radially opposite on scale No. 2, and can be read by the use of either the fixed or movable finger, as happens to be most convenient. Thus, on scale 2, the movable pointer C indicates 135, and on scale 1 we have the corresponding common logarithm ·13 without the characteristic, which depends upon the position of the decimal point in the 135. The characteristic of the logarithm is found in the usual way, and must be added by the operator just the same as when working from a book of tables.

Multiplication.—Use scale No. 2 and proceed as follows:

1. Set one factor under the pointer D by turning nut A.
2. Set movable pointer C at 1 by turning nut B.
3. Turn nut A till second factor comes under pointer C.
4. Read off result under pointer D.

Division.—Use scale No. 2 and proceed as follows:

1. Set the dividend under the pointer D.
2. Set the pointer C to the divisor by turning nut B.
3. Turn dial, by nut A, till 1 comes under pointer C.
4. Read off result under pointer D.

Square Roots.—Use scales Nos. 3 and 4. Although there are two circles, one is really a continuation of the other. The numbers on scale No. 2 are the squares of those on scales 3 and 4. To find square root of a number proceed as follows:

1. Set the number on scale No. 2, under one of the pointers (whichever is convenient).
2. Ascertain whether the number (neglecting decimal figures, if any) is even or odd.
3. If number is even, read off square root on scale No. 3.
4. If number is odd, read off square root on scale No. 4.

Thus, the pointer C indicates 135 on scale 2. If the decimal point comes after the 5, there are three (an odd number of figures). Hence look on scale 4 and we find 11·6 as the square root. The decimal point is put in by the operator. If the original number were 13·5 (an even number of figures) look on scale 3, and the square root is given as 3·68.

Squares.—Use scales No. 3 and 4, and No. 2, and proceed as follows:—

1. Set one of the pointers over the number on scale No. 3 or 4, whichever the number is on (scale No. 3 being simply a continuation of No. 4)
2. Read off the number radially opposite, on scale 2. This is the square.

Sine of an Angle.—Use scale No. 5 and scale No. 1, and proceed as follows:—

1. Set one of the pointers over the angle on scale No. 5.
2. Read off value of the sine radially opposite on scale No. 1.

Cosine of an Angle.—This can be readily deduced from scale No. 5, since the cosine of an angle is the sine of its complement. For example, the cosine of 60 deg. is the same as the sine of 30 deg.

Cubes and Cube Roots, and Miscellaneous Powers Generally.—These can be obtained by means of scales Nos. 1 and 2 just as readily as by the aid of a table of logarithms.

Hyperbolic Logarithms.—These may be found by multiplying the common logarithm on scale No. 1 by 2·30, which point is marked for convenience of reference on scale No. 2.

Note.—In setting the pointers of the instrument to fractions of a division the judgment of the operator must be used, and the same remark applies to the reading of the results. The position of the decimal place can be determined by inspection without specific rules. With a very little practice the operator will find it possible to obtain results accurate to less than one per cent, which is closer than is required in most engineering calculations.

Fractions.—In working out long sets of fractions many operations and much tedious labour can be saved by the use of the instrument, as may be seen from the following example:—

Required the value of $\frac{29{\cdot}7 \times 43{\cdot}5 \times 82{\cdot}9}{2{\cdot}7 \times 5{\cdot}72}$

This can be obtained by the instrument in the following way, using scale No. 2.

1. Turn nut A till 29·7 comes under the fixed pointer D.
2. Turn nut B till pointer C is opposite the first divisor 2·7.
3. Turn nut A till 43·5 comes opposite movable pointer C.
4. Turn nut B till movable pointer C is opposite 5·72.
5. Turn nut A till 82·9 is opposite movable pointer C.
6. Read off result 695 behind the fixed pointer D.

The actual result by inspection can be seen to be approximately 7000, so the interpretation of the reading may without hesitation be written 6950.

The general rule for dealing with fractional calculations is as follows: Put the sum to be worked in the form $\frac{a \times b \times c \times \ldots\ldots}{m \times n \times \ldots\ldots}$ and disregard the decimal point in the result until it is reached, then proceed as follows:—

1. Set the dial by nut A till first multiplier (a) comes behind the fixed pointer D.
2. Turn the movable pointer C to the first divisor (m).
3. Turn the dial by nut A till second multiplier (b) comes behind movable pointer C.
4. Turn the movable pointer C to the second divisor (n), and so on till all the numbers are used up.
5. Read off result behind fixed pointer D, and estimate position of decimal place.

N.B.—If there are not enough divisions or multipliers in the fraction, use the number 1 instead as often as may be required, that is, until there is one more factor in the numerator than in the denominator.

Thus $\frac{a \times b}{m \times n}$ would be worked as $\frac{a \times b \times 1}{m \times n}$

$\frac{a \times b}{m \times n \times p}$ would be worked as $\frac{a \times b \times 1 \times 1}{m \times n \times p}$

$\frac{a \times b \times c \times d}{m \times n}$ would be worked as $\frac{a \times b \times c \times d}{m \times n}$

The fixed pointer is only used to set instrument at first, and read results at finish.

The movable pointer is shifted only for divisors.

The dial is shifted only for multipliers.

Those desirous of mastering the logarithmic principles on which the working of the instrument is based will find a full description in *The Mechanical Engineer* for July 23, 1898. The instrument can, of course, be used with perfect accuracy without any knowledge of logarithms, but an intelligent application of the scientific principles of any arrangement always enables the operator to use it with greater confidence.

APPENDIX.

Professor Ripper's Mean Pressure Indicator. — This instrument indicates the mean forward and back pressures on the piston of an engine continuously, the respective pressures being shown by a pair of ordinary pressure gauges. Each end of the cylinder is in turn placed in communication with a pressure gauge, so that one gauge continuously indicates back pressure, and the other gauge the forward pressure.

The form of instrument shown in Fig. 260 is arranged for a vertical cylinder, though the instrument for a horizontal cylinder is practically the same. The left-hand view is a part section looking directly through the instrument at right angles to the axis of the cylinder, and the right-hand view is at right angles to it.

Slightly modified ordinary indicator cocks are screwed into the cylinder at C and D, and attached to each cock is a rotating valve in a valve box at E and F. These valve boxes are connected together by a pair of parallel pipes, each of which communicates with its pressure gauge.

Fig. 261 is an enlarged section of one of the valve boxes, showing the valve in section with its ports.

It will be noticed that the valve is hollow, and contains ports D D, which open the valve chamber C alternately to the pipes F and E, thus putting them, in turn, in communication with the channel A leading to that indicator cock.

The valve is rotated by the chain wheel G, which is driven from the crankshaft of the engine to be indicated.

It will be now seen that as the rotating valves open and close their respective ports at the end of a stroke one end of the cylinder will be in communication with one pressure gauge, and the other end with the second gauge for the whole of one stroke, at the end of which the two ends of the cylinder are switched over to the opposite gauges. In this way one gauge always records forward pressure, and the other back pressure.

The variation of pressure in the engine cylinder is not communicated to the pressure gauges on account of the small channels and throttling cocks, one in each fitting

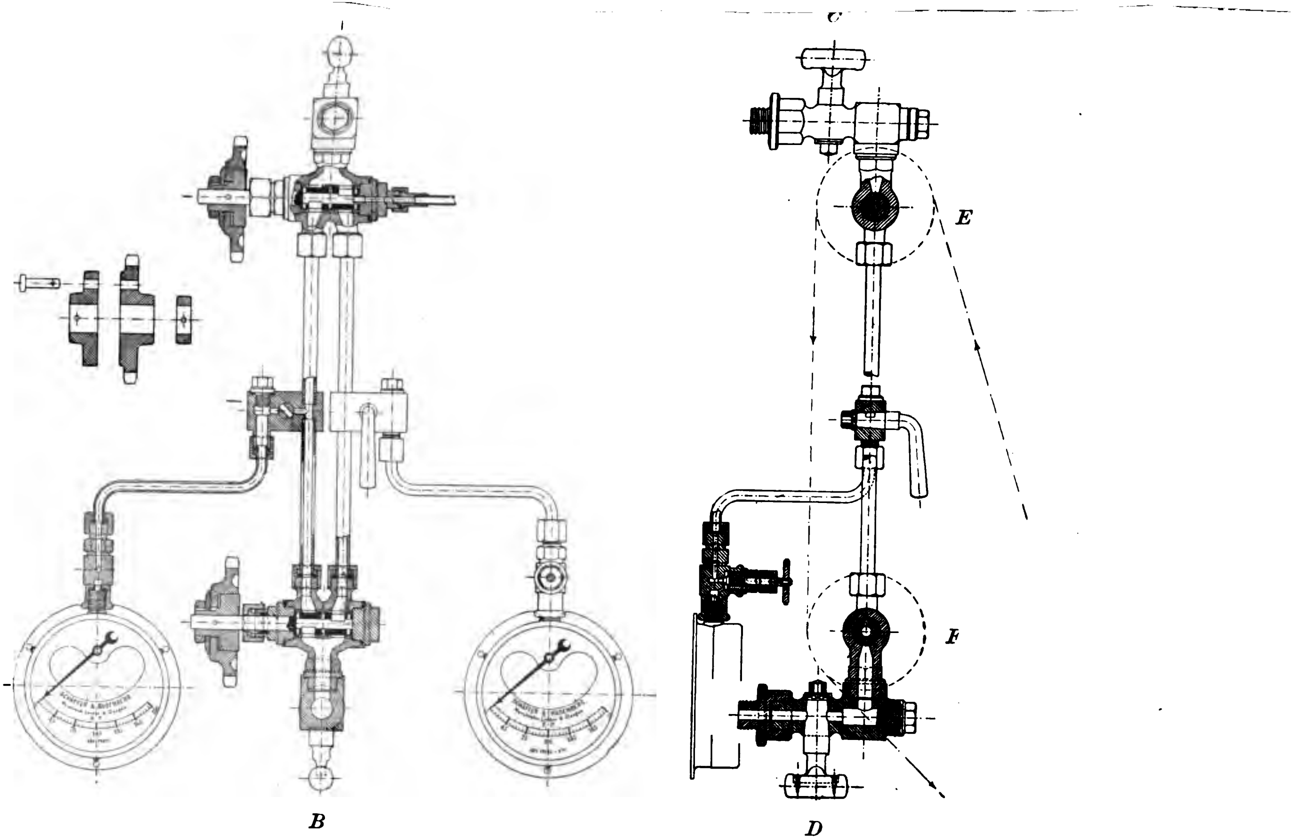


FIG. 260.—Professor Ripper's Mean Pressure Indicator.

half way down the cylinder, and another attached to the pressure gauge. In this manner the steam and water is so throttled that the gauge pointer can be made to indicate steadily the mean forward or back pressure in the cylinder. The average is a *time* average, but this only

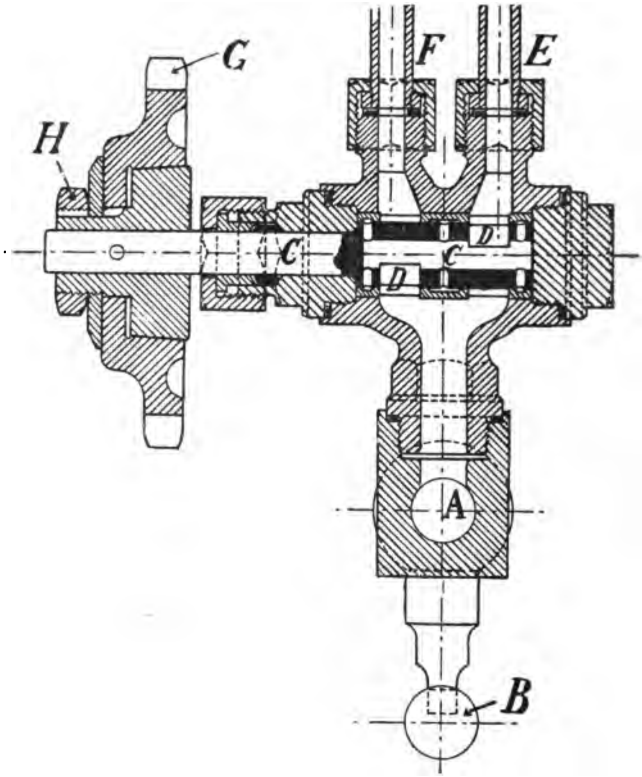


FIG. 261.—Section through Valve Box of Professor Ripper's Indicator.

differs from a space average by a very small amount, generally three per cent, which can be allowed for, or a second graduation can be made on the gauge dial.

It is imperative that the pipe leading from the first throttling cock to a pressure gauge shall be always full, or nearly full of water, hence the position of the gauges.

In Fig. 261 the chain wheel G is connected to the spindle through a conical friction clutch held in gear by

the nut H. This permits of easy adjustment of the valve at the time of fixing.

The Mechanical Equivalent of Heat.—Until quite recently Joule's value of the mechanical equivalent of heat, namely, 772 foot-pounds of energy per British thermal unit, has been used. Prof. Rowland, in America, re-determined the mechanical equivalent on a more extended scale than Joule, and over a much more extended range of temperatures; and the number now used as determined by him is 778 foot-pounds, or 426·9 meter-kilograms, when the thermal unit is the calorie; that is, one kilogram of water raised 1 deg. C.

Mr. Moorby, in conjunction with Professor Osborne Reynolds, has recently re-determined Joule's equivalent, at Manchester, with a Froude dynamometer brake, and has found 777 as its value when the range of temperature of the water acted upon was from freezing to boiling points.

As the specific heat of water varies, the mechanical equivalent of heat must, of necessity, vary according to the mean temperature of the water during the experiment. It thus becomes necessary to state distinctly at what standard temperature the mechanical equivalent is determined. For convenience this standard temperature is taken as 60 deg. Fah.

STEAM TABLE.

Absolute Pressure in Pounds per square inch.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.*, B. T. U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.*, B. T. U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
1	102·02	70·04	1043·02	1113·06	61·62	330·4	·00303
2	126·30	94·37	1026·09	1120·46	64·11	171·9	·00582
3	141·65	109·76	1015·38	1125·14	65·65	117·3	·00852
4	153·12	121·27	1007·37	1128·64	66·77	89·5	·01117
5	162·37	130·56	1000·90	1131·46	67·66	72·56	·01378
6	170·17	138·40	995·44	1133·84	68·40	61·14	·01636
7	176·94	145·21	990·69	1135·91	69·04	52·89	·01891
8	182·95	151·25	986·48	1137·74	69·60	46·65	·02144
9	188·36	156·70	982·69	1139·39	70·11	41·77	·02394
10	193·28	161·66	979·23	1140·89	70·56	37·83	·02644
11	197·81	166·23	976·05	1142·27	70·97	34·59	·02891
12	202·01	170·46	973·10	1143·55	71·33	31·87	·03138
13	205·93	174·40	970·35	1144·75	71·66	29·56	·03383
14	209·60	178·11	967·76	1145·87	71·97	27·58	·03626
14·7	212·00	180·53	966·07	1146·60	72·18	26·37	·03793
15	213·07	181·61	965·32	1146·93	72·27	25·85	·03869
16	216·35	184·92	963·01	1147·93	72·55	24·33	·04111
17	219·45	188·06	960·82	1148·87	72·81	22·98	·04352
18	222·42	191·06	958·72	1149·78	73·06	21·78	·04592
19	225·26	193·92	956·73	1150·64	73·30	20·70	·04831
20	227·96	196·66	954·81	1151·47	73·52	19·73	·05069
21	230·57	199·29	952·98	1152·26	73·74	18·84	·05307
22	233·07	201·82	951·21	1153·03	73·94	18·04	·05545
23	235·48	204·26	949·50	1153·76	74·14	17·30	·05781
24	237·80	206·61	947·86	1154·47	74·32	16·62	·06017
25	240·05	208·89	946·27	1155·16	74·50	16·00	·06252
26	242·23	211·09	944·73	1155·82	74·68	15·42	·06487
27	244·33	213·22	943·24	1156·46	74·85	14·88	·06721
28	246·38	215·29	941·79	1157·08	75·01	14·38	·06955
29	248·36	217·31	940·38	1157·69	75·17	13·91	·07187
30	250·29	219·26	939·02	1158·28	75·32	13·48	·07420

Absolute Pressure in Pounds per square inch.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, i.e., B. T. U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, i.e., B. T. U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
31	252·17	221·17	937·69	1158·85	75·47	13·07	·07652
32	254·00	223·02	936·39	1159·41	75·61	12·68	·07884
33	255·78	224·83	935·13	1159·95	75·74	12·32	·08115
34	257·52	226·59	933·89	1160·48	75·88	11·98	·08346
35	259·22	228·32	932·69	1161·00	76·01	11·66	·08577
36	260·88	230·00	931·51	1161·51	76·13	11·36	·08807
37	262·51	231·65	930·35	1162·00	76·26	11·07	·09036
38	264·09	233·26	929·23	1162·49	76·38	10·79	·09266
39	265·65	234·84	928·12	1162·96	76·49	10·53	·09495
40	267·17	236·39	927·04	1163·43	76·61	10·28	·09723
41	268·66	237·90	925·98	1163·88	76·72	10·05	·09951
42	270·12	239·39	924·94	1164·33	76·83	9·83	·10179
43	271·56	240·85	923·92	1164·77	76·93	9·61	·10407
44	272·97	242·28	922·92	1165·19	77·04	9·40	·10635
45	274·35	243·68	921·94	1165·62	77·14	9·21	·10862
46	275·70	245·06	920·97	1166·03	77·24	9·02	·11088
47	277·04	246·42	920·02	1166·44	77·33	8·84	·11315
48	278·35	247·75	919·08	1166·84	77·43	8·67	·11541
49	279·64	249·06	918·16	1167·23	77·52	8·50	·11767
50	280·90	250·36	917·26	1167·62	77·61	8·34	·11993
51	282·15	251·62	916·37	1167·99	77·69	8·19	·12218
52	283·38	252·87	915·49	1168·37	77·78	8·04	·12443
53	284·59	254·11	914·63	1168·74	77·87	7·89	·12668
54	285·78	255·32	913·78	1169·10	77·95	7·76	·12893
55	286·95	256·52	912·94	1169·46	78·04	7·62	·13117
56	288·11	257·69	912·12	1169·81	78·12	7·49	·13341
57	289·25	258·86	911·30	1170·16	78·20	7·37	·13565
58	290·37	260·00	910·50	1170·50	78·27	7·25	·13789
59	291·48	261·13	909·71	1170·84	78·35	7·14	·14013
60	292·57	262·25	908·93	1171·18	78·42	7·02	·14236

Absolute Pressure in Pounds per square inch.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.*, B. T. U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.*, B. T. U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
61	293·65	263·35	908·16	1171·51	78·49	6·92	·14459
62	294·72	264·43	907 39	1171·83	78·57	6·81	·14682
63	295·77	265·51	906·64	1172·15	78·64	6·71	·14905
64	296·81	266·57	905·90	1172·47	78·71	6·61	·15128
65	297·83	267·61	905·17	1172·78	78·78	6·52	·15350
66	298·84	268·64	904·44	1173·09	78·85	6·42	·15572
67	299·84	269·67	903·73	1173·39	78·91	6·33	·15794
68	300·83	270·67	903·02	1173·69	78·98	6·24	·16016
69	301·81	271·67	902 32	1173·99	79·04	6·16	·16237
70	302·77	272·66	901·63	1174·29	79·11	6·08	·16458
71	303·73	273·63	900·95	1174·58	79·17	5·99	·16679
72	304·67	274·60	900·27	1174·87	79·23	5·92	·16900
73	305·60	275·55	899·60	1175·15	79·29	5·84	·17121
74	306·53	276·49	898·94	1175·43	79·35	5·77	·17342
75	307·44	277·43	898·28	1175·71	79·41	5·69	·17562
76	308·34	278·35	897·64	1175·99	79·47	5·62	·17783
77	309·24	279 27	896·99	1176·26	79·53	5·56	·18003
78	310·12	280·17	896·36	1176·53	79·58	5·49	·18223
79	311·00	281·07	895·73	1176·80	79·64	5·42	·18443
80	311·87	281·95	895·11	1177·06	79·69	5·36	·18663
81	312·73	282·83	894·49	1177·32	79·75	5·30	·18882
82	313·58	283·70	893·88	1177·58	79·80	5·24	·19102
83	314·42	284·56	893·28	1177·84	79·86	5·18	·19321
84	315·25	285·41	892 68	1178·09	79·91	5·12	·19540
85	316·08	286·26	892·08	1178·34	79·96	5·06	·19759
86	316·89	287·10	891·50	1178·59	80·01	5·01	·19078
87	317·71	287·93	890·91	1178·84	80·06	4·95	·20197
88	318·51	288·75	890·34	1179·09	80·11	4·90	·20416
89	319·31	289·57	889·76	1179·33	80·16	4·85	·20634
90	320·09	290·37	889·20	1179·57	80·21	4·80	·20853

Absolute Pressure in Pounds per square inch.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.*, B. T. U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.*, B. T. U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
91	320·88	291·18	888·63	1179·81	80·26	4·75	·21071
92	321·65	291·97	888·08	1180·05	80·31	4·70	·21289
93	322·42	292·76	887·52	1180·28	80·35	4·65	·21507
94	323·18	293·54	886·97	1180·51	80·40	4·60	·21725
95	323·94	294·31	886·43	1180·74	80·44	4·56	·21943
96	324·69	295·08	885·89	1180·97	80·49	4·51	·22160
97	325·43	295·85	885·35	1181·20	80·53	4·47	·22378
98	326·17	296·60	884·82	1181·42	80·58	4·43	·22595
99	326·90	297·35	884·30	1181·65	80·62	4·38	·22812
100	327·63	298·09	883·77	1181·87	80·67	4·34	·23029
101	328·35	298·83	883·25	1182·09	80·71	4·30	·23246
102	329·06	299·57	882·74	1182·30	80·75	4·26	·23463
103	329·77	300·29	882·23	1182·52	80·79	4·22	·23680
104	330·47	301·01	881·72	1182·73	80·84	4·19	·23897
105	331·17	301·73	881·21	1182·95	80·88	4·15	·24114
106	331·86	302·44	880·71	1183·16	80·92	4·11	·24330
107	332·55	303·15	880·21	1183·37	80·96	4·07	·24547
108	333·23	303·85	879·72	1183·57	80·99	4·04	·24763
109	333·91	304·55	879·23	1183·78	81·03	4·00	·24979
110	334·58	305·24	878·74	1183·99	81·07	3·97	·25195
111	335·25	305·93	878·26	1184·19	81·11	3·93	·25411
112	335·91	306·61	877·78	1184·39	81·15	3·90	·25626
113	336·57	307·29	877·31	1184·59	81·18	3·87	·25842
114	337·23	307·96	876·84	1184·79	81·22	3·84	26058
115	337·87	308·62	876·37	1184·99	81·26	3·81	·26273
116	338·52	309·28	875·91	1185·19	81·29	3·78	·26489
117	339·16	309·94	875·44	1185·38	81·33	3·75	·26704
118	339·80	310·59	874·99	1185·58	81·37	3·72	·26919
119	340·43	311·24	874·53	1185·77	81·40	3·69	·27135
120	341·06	311·89	874·08	1185·96	81·44	3·66	·27350

Absolute Pressure in Pounds per square inch.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.*, B. T. U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.*, B. T. U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units=Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
121	341·68	312·52	873·63	1186·15	81·47	3·63	·27565
122	342·30	313·16	873·18	1186·34	81·51	3·60	·27780
123	342·92	313·79	872·73	1186·53	81·54	3·57	·27995
124	343·53	314·42	872·29	1186·71	81·58	3·55	·28210
125	344·14	315·05	871·85	1186·90	81·61	3·52	·28424
126	344·74	315·67	871·41	1187·08	81·65	3·49	·28639
127	345·34	316·29	870·98	1187·27	81·68	3·47	·28853
128	345·94	316·90	870·55	1187·45	81·71	3·44	·29068
129	346·53	317·51	870·12	1187·63	81·74	3·42	·29282
130	347·12	318·12	869·69	1187·81	81·77	3·39	·29496
131	347·71	318·73	869·26	1187·99	81·81	3·37	·29710
132	348·29	319·33	868·84	1188·17	81·84	3·34	·29924
133	348·87	319·92	868·42	1188·34	81·87	3·32	·30138
134	349·44	320·52	868·01	1188·52	81·90	3·29	·30352
135	350·02	321·11	867·59	1188·69	81·93	3·27	·30566
136	350·58	321·69	867·18	1188·87	81·96	3·25	·30780
137	351·15	322·27	866·77	1189·04	81·99	3·23	·30993
138	351·71	322·85	866·36	1189·21	82·02	3·20	·31207
139	352·27	323·43	865·96	1189·38	82·05	3·18	·31420
140	352·83	324·00	865·55	1189·56	82·08	3·16	·31634
141	353·38	324·57	865·15	1189·72	82·11	3·14	·31847
142	353·93	325·14	864·75	1189·89	82·14	3·12	·32060
143	354·48	325·71	864·35	1190·06	82·17	3·10	·32273
144	355·02	326·27	863·96	1190·23	82·19	3·08	·32487
145	355·56	326·82	863·57	1190·39	82·22	3·06	·32700
146	356·10	327·38	863·18	1190·55	82·25	3·04	·32913
147	356·64	327·93	862·79	1190·72	82·28	3·02	·33126
148	357·17	328·48	862·40	1190·88	82·30	3·00	·33339
149	357·70	329·02	862·02	1191·04	82·33	2·98	·33552
150	358·22	329·57	861·63	1191·20	82·36	2·96	·33764

Absolute Pressure in Pounds per square inch.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.*, B. T. U., required to raise the Temperature of 1 lb. Water from 32° to *t*° F.	Latent Heat, *i.e.*, B. T. U., required to convert 1 lb. Water at *t*° into Steam at *t*°.	Total Heat of Evaporation from 32° F. and at *t*° F. in British Thermal Units=Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
151	358·62	330·12	861·23	1191·36	82·38	2·94	·33986
152	359·14	330·63	860·85	1191·51	82·41	2·93	·34199
153	359·66	331·18	860·47	1191·67	82·43	2·91	·34411
154	360·17	331·72	860·10	1191·82	82·46	2·89	·34624
155	360·68	332·24	859·72	1191·98	82·48	2·88	·34836
156	361·20	332·77	859·35	1192·13	82·51	2·86	·35049
157	361·70	333·30	858·98	1192·28	82·53	2·84	·35261
158	362·21	333·82	858·61	1192·44	82·56	2·83	·35474
159	362·70	334·34	858·24	1192·61	82·59	2·81	·35686
160	363·34	334·85	857·91	1192·76	82·62	2·79	·35889
161	363·63	335·38	857·54	1192·91	82·64	2·78	·36101
162	364·20	335·89	857·16	1193·06	82·66	2·76	·36313
163	364·69	336·40	856·80	1193·21	82·69	2·75	·36524
164	365·18	336·91	856·44	1193·36	82·71	2·73	·36736
165	365·68	337·41	856·09	1193·50	82·73	2·72	·36947
166	366·17	337·92	855·71	1193·65	82·75	2·70	·37160
167	366·65	338·41	855·36	1193·80	82·78	2·68	·37372
168	367·13	338·90	855·00	1193·95	82·80	2·67	·37583
169	367·62	339·41	854·66	1194·10	82·82	2·65	·37795
170	368·23	339·89	854·36	1194·25	82·85	2·63	·38007
171	368·59	340·38	853·95	1194·39	82·87	2·62	·38219
172	369·04	340·87	853·61	1194·53	82·89	2·60	·38430
173	369·51	341·35	853·28	1194·67	82·92	2·59	·38641
174	369·98	341·84	852·13	1194·82	82·94	2·57	·38853
175	370·45	342·33	852·60	1194·96	82·96	2·56	·39064
176	370·90	342·80	852·27	1195·10	82·98	2·54	·39274
177	371·37	343·29	851·93	1195·24	83·00	2·53	·39486
178	371·83	343·76	851·60	1195·38	83·03	2·51	·39697
179	372·29	344·23	851·27	1195·52	83·05	2·50	·39908
180	372·89	344·71	850·96	1195·67	83·07	2·49	·40120

Absolute Pressure in Pounds per square inch.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.*, B. T. U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.*, B. T. U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
181	373·20	345·18	850·54	1195·80	83·09	2·48	·40331
182	373·66	345·65	850·22	1195·94	83·11	2·47	·40542
183	374·11	346·12	849·89	1196·08	83·13	2·45	·40754
184	374·56	346·58	849·57	1196·22	83·15	2·44	·40965
185	375·01	347·05	849·26	1196·36	83·17	2·43	·41175
186	375·45	347·51	848·95	1196·48	83·19	2·42	·41386
187	375·90	347·97	848·63	1196·62	83·21	2·40	·41596
188	376·34	348·42	848·30	1196·76	83·23	2·39	·41807
189	376·78	348·87	847·99	1196·90	83·25	2·38	·42020
190	377·35	349·33	847·70	1197·03	83·27	2·37	·42228
191	377·22	349·77	847·38	1197·16	83·29	2·36	·42438
192	378·09	350·22	847·06	1197·29	83·31	2·35	·42648
193	378·52	350·67	846·75	1197·42	83·32	2·34	·42859
194	378·95	351·12	846·43	1197·55	83·34	2·33	·43069
195	379·38	351·57	846·12	1197·68	83·35	2·31	·43279
196	379·90	352·02	845·80	1197·81	83·37	2·30	·43490
197	380·23	352·45	845·50	1197·94	83·39	2·29	·43700
198	380·65	352·89	845·19	1198·07	83·41	2·28	·43910
199	381·08	353·33	844·87	1198·20	83·43	2·27	·44121
200	381·64	353·77	844·57	1198·34	83·46	2·26	·44331
201	382·1	354·1	844·3	1198·46	83·48	2·250	·4453
202	382·5	354·6	844·0	1198·59	83·50	2·236	·4475
203	382·9	355	843·7	1198·71	83·51	2·227	·4496
204	383·3	355·3	843·3	1198·83	83·53	2·217	·4516
205	383·7	355·8	843·0	1198·96	83·55	2·206	·4538
206	384·1	356·3	842·7	1199·09	83·57	2·196	·4558
207	384·5	356·8	842·4	1199·21	83·59	2·186	·4580
208	384·9	357·2	842·1	1199·33	83·61	2·176	·4600
209	385·3	357·7	841·8	1199·46	83·63	2·166	·4621
210	385·7	358·1	841·5	1199·57	83·65	2·155	·4644

Absolute Pressure in Pounds per square inch.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.*, B. T. U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.*, B. T. U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
211	386·1	358·6	841·2	1199·71	83·66	2·146	·4664
212	386·5	359	840·9	1199·83	83·68	2·137	·4684
213	386·9	359·4	840·6	1199·95	83·70	2·128	·4706
214	387·3	359·9	840·3	1200·08	83·72	2·119	·4726
215	387·7	360·2	840·0	1200·20	83·73	2·109	·4746
216	388·0	360·6	839·8	1200·32	83·75	2·100	·4766
217	388·4	361·1	839·5	1200·45	83·77	2·090	·4787
218	388·8	361·5	839·2	1200·57	83·78	2·081	·4808
219	389·3	361·9	838·9	1200·69	83·80	2·072	·4830
220	389·7	362·3	838·6	1200·82	83·82	2·062	·4850
221	390·1	362·7	838·3	1200·95	83·84	2·053	·4870
222	390·5	363·1	838·0	1201·07	83·85	2·045	·4891
223	390·9	363·5	837·8	1201·19	83·87	2·036	·4912
224	391·3	363·9	837·5	1201·30	83·88	2·028	·4934
225	391·6	364·3	837·2	1201·43	83·89	2·020	·4956
226	392·0	364·8	836·9	1201·55	83·91	2·011	·4977
227	392·4	365·1	836·6	1201·66	83·93	2·002	·4999
228	392·8	365·5	836·3	1201·77	83·94	1·994	·5020
229	393·2	365·9	836·0	1201·89	83·96	1·985	·5040
230	393·6	366·3	835·8	1202·01	83·97	1·976	·5062
231	394·0	366·7	835·5	1202·11	83·99	1·968	·5082
232	394·3	367·1	835·3	1202·22	84·00	1·960	·5103
233	394·7	367·5	835·0	1202·33	84·01	1·952	·5124
234	395·1	367·9	834·8	1202·44	84·03	1·944	·5145
235	395·5	368·2	834·5	1202·55	84·04	1·936	·5166
236	395·9	368·6	834·3	1202·66	84·06	1·928	·5186
237	396·3	369	834·0	1202·77	84·08	1·920	·5208
238	396·6	369·4	833·7	1202·88	84·09	1·910	·5228
239	397·0	369·8	833·4	1202·99	84·10	1·904	·5249
240	397·4	370·1	833·1	1203·10	84·12	1·897	·5270

Absolute Pressure in Pounds per square inch.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.*, B. T. U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.*, B. T. U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
241	397·8	370·5	832·8	1203·21	84·13	1·889	·5290
242	398·1	370·9	832·6	1203·32	84·15	1·882	·5310
243	398·5	371·3	832·3	1203·43	84·17	1·875	·5330
244	398·9	371·7	832·1	1203·53	84·18	1·868	·5351
245	399·2	372·1	831·8	1203·66	84·20	1·861	·5372
246	399·6	372·4	831·5	1203·77	84·21	1·853	·5393
247	400·0	372·8	831·3	1203·88	84·22	1·847	·5414
248	400·3	373·2	831·0	1203·00	84·24	1·839	·5435
249	400·7	373·6	830·7	1204·10	84·25	1·832	·5455
250	401·1	374·0	830·5	1204·21	84·27	1·825	·5476
251	401·4	374·3	830·2	1204·32	84·29	1·818	·5497
252	401·7	374·7	830·0	1204·43	84·30	1·812	·5517
253	402·1	375·0	829·8	1204·54	84·31	1·805	·5538
254	402·4	375·3	829·5	1204·65	84·32	1·798	·5559
255	402·7	375·7	829·2	1204·75	84·33	1·792	·5580
256	403·1	376·0	828·9	1204·86	84·34	1·785	·5601
257	403·5	376·3	828·6	1204·97	84·35	1·778	·5622
258	403·9	376·7	828·3	1205·08	84·36	1·772	·5643
259	404·2	377·1	828·0	1205·19	84·38	1·765	·5664
260	404·6	377·4	827·8	1205·29	84·39	1·759	·5685
261	404·9	377·8	827·6	12053·9	84·40	1·752	·5705
262	405·2	378·2	827·3	1205·50	84·42	1·747	·5725
263	405·5	378·5	827·1	1205·60	84·43	1·740	·5746
264	405·8	378·8	826·8	1205·70	84·44	1·733	·5767
265	406·1	379·2	826·6	1205·80	84·45	1·728	·5788
266	406·5	379·6	826·3	1205·90	84·47	1·721	·5810
267	406·9	379·9	826·0	1206·00	84·48	1·716	·5830
268	407·2	380·2	825·7	1206·11	84·49	1·710	·5851
269	407·5	380·6	825·5	1206·21	84·50	1·703	·5872
270	407·8	381·0	825·3	1206·31	84·51	1·697	·5894

Absolute Pressure in Pounds per square inch.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.*, B. T. U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.*, B. T. U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
271	408·1	381·3	825·0	1206·41	84·52	1·691	·5915
272	408·4	381·7	824·8	1206·51	84·54	1·685	·5935
273	408·8	382·0	824·6	1206·61	84·55	1·680	·5956
274	409·1	382·3	824·3	1206·71	84·56	1·674	·5978
275	409·4	382·6	824·0	1206·81	84·58	1·668	·5999
276	409·8	383·0	823·8	1206·91	84·59	1·663	·6020
277	410·0	383·3	823·6	1207·02	84·60	1·657	·6040
278	410·4	383·6	823·4	1207·12	84·61	1·650	·4060
279	410·8	384·0	823·1	1207·22	84·62	1·645	·6080
280	411·1	384·3	822·9	1207·32	84·63	1·639	·6101
281	411·4	384·7	822·7	1207·41	84·64	1·633	·6122
282	411·8	385·0	822·5	1207·51	84·66	1·628	·6143
283	412·1	385·3	822·2	1207·61	84·67	1·623	·6164
284	412·4	385·6	822·0	1207·70	84·68	1·617	·6185
285	412·7	386·0	821·7	1207·80	84·69	1·612	·6206
286	413·0	386·3	821·5	1207·90	84·70	1·606	·6227
287	413·4	386·6	821·3	1208·00	84·71	1·601	·6248
288	413·7	387·0	821·0	1208·10	84·72	1·596	·6269
289	414·0	387·3	820·8	1208·20	84·73	1·590	·6290
290	414·3	387·6	820·6	1208·30	84·74	1·585	·6310
291	414·6	387·9	820·3	1208·40	84·75	1·580	·6330
292	415·0	388·2	820·1	1208·49	84·76	1·575	·6351
293	415·3	388·5	819·9	1208·59	84·77	1·570	·6371
294	415·6	388·8	819·7	1208·68	84·78	1·565	·6392
295	415·9	389·2	819·5	1208·77	84·79	1·560	·6414
296	416·2	389·5	819·2	1208·86	84·80	1·555	·6434
297	416·6	389·8	819·0	1208·95	84·81	1·550	·6455
298	416·9	390·1	818·8	1209·06	84·82	1·545	·6475
299	417·2	390·4	818·6	1209·15	84·83	1·540	·6495
300	417·5	390·8	818·4	1209·25	84·84	1·536	·6515

STEAM.

TEMPERATURE—PRESSURE TABLE.

Temp. F.	Pressure.	Temp. F.	Pressure.	Temp. F.	Pressure.	Temp. F.	Pressure.
60	·26	95	·81	130	2·21	165	5·3
61	·26	96	·83	131	2·27	166	5·5
62	·27	97	·86	132	2·33	167	5·6
63	·28	98	·89	133	2·40	168	5·7
64	·29	99	·91	134	2·46	169	5·9
65	·30			135	2·52		
66	·31	100	·94	136	2·59	170	6·0
67	·32	101	·97	137	2·66	171	6·1
68	·33	102	1·00	138	2·73	172	6·3
69	·35	103	1·03	139	2·80	173	6·4
		104	1·06			174	6·6
70	·36	105	1·09	140	2·88	175	6·7
71	·37	106	1·13	141	2·95	176	6·9
72	·38	107	1·16	142	3·03	177	7·0
73	·40	108	1·19	143	3·11	178	7·2
74	·41	109	1·23	144	3·19	179	7·3
75	·42			145	3·27		
76	·44	110	1·27	146	3·35	180	7·5
77	·45	111	1·30	147	3·44	181	7·7
78	·47	112	1·34	148	3·53	182	7·8
79	·49	113	1·38	149	3·62	183	8·0
		114	1·42			184	8·2
80	·50	115	1·46	150	3·7	185	8·4
81	·52	116	1·50	151	3·8	186	8·6
82	·53	117	1·55	152	3·9	187	8·8
83	·55	118	1·59	153	4·0	188	8·9
84	·57	119	1·64	154	4·1	189	9·1
85	·59			155	4·2		
86	·61	120	1·68	156	4·3	190	9·3
87	·63	121	1·73	157	4·4	191	9·5
88	·65	122	1·78	158	4·5	192	9·7
89	·67	123	1·83	159	4·6	193	10·0
		124	1·88			194	10·2
90	·69	125	1·93	160	4·7	195	10·4
91	·71	126	1·98	161	4·8	196	10·6
92	·74	127	2·04	162	5·0	197	10·8
93	·76	128	2·10	163	5·1	198	11·1
94	·78	129	2·15	164	5·2	199	11·3

Temp. F.	Pressure.	Temp. F.	Pressure.	Temp. F.	Pressure.	Temp. F.	Pressure.
200	11·5	237	23·7	273	44·1	310	77·9
201	11·8	238	24·1	274	44·8	311	79·1
202	12·0	239	24·6	275	45·5	312	80·2
203	12·3			276	46·3	313	81·4
204	12·5	240	25·0	277	47·0	314	82·6
205	12·8	241	25·5	278	47·8	315	83·8
206	13·0	242	25·9	279	48·6	316	85·0
207	13·3	243	26·4			317	86·2
208	13·6	244	26·9	280	49·3	318	87·5
209	13·8	245	27·4	281	50·1	319	88·7
		246	27·8	282	50·9		
210	14·1	247	28·3	283	51·7	320	90·0
211	14·4	248	28·9	284	52·6	321	91·3
212	14·7	249	29·4	285	53·4	322	92·5
213	15·0			286	54·2	323	93·9
214	15·3	250	29·9	287	55·1	324	95·2
215	15·6	251	30·4	288	56·0	325	99·5
216	15·9	252	30·9	289	56·8	326	97·9
217	16·2	253	31·5			327	99·2
218	16·5	254	32·0	290	57·7	328	100·6
219	16·9	255	32·6	291	58·6	329	102·0
		256	33·2	292	59·5		
220	17·2	257	33·7	293	60·5	330	103·4
221	17·5	258	34·3	294	61·4	331	104·9
222	17·9	259	34·9	295	62·3	332	106·3
223	18·2			296	63·3	333	107·8
224	18·6	260	35·5	297	64·3	334	109·3
225	18·9	261	36·1	298	65·2	335	110·7
226	19·3	262	36·7	299	66·2	336	112·2
227	19·7	263	37·4			337	113·8
228	20·0	264	38·0	300	67·2	338	115·3
229	20·4	265	38·6	301	68·2	339	116·9
		266	39·3	302	69·3		
230	20·8	267	39·9	303	70·3	340	118·4
231	21·2	268	40·6	304	71·4	341	120·0
232	21·6	269	41·3	305	72·4	342	121·6
233	22·0			306	73·5	343	123·3
234	22·4	270	42·0	307	74·6	344	124·9
235	22·8	271	42·7	308	75·7	345	126·6
236	23·3	272	43·4	309	76·8	346	128·2

emp. F.	Pressure.	Temp F.	Pressure.	Temp. F.	Pressure.	Temp. F.	Pressure.
347	29·3	373	180·4	400	247·8	427	332·8
348	131·6	374	182·6	401	250·6	428	336·4
349	133·4	375	184·9	402	253·4	429	339·9
		376	187·1	403	256·3		
350	135·1	377	189·4	404	259·2	430	343·5
351	136·9	378	191·7	405	262·1	431	347·1
352	138·7	379	194·0	406	265·0	432	350·7
353	140·5			407	268·0	433	354·4
354	142·3	380	196·3	408	271·0	434	358·0
355	144·1	381	198·7	409	274·0	435	361·7
356	146·0	382	201·1			436	365·5
357	147·8	383	203·5	410	277·1	437	369·2
358	149·7	384	205·9	411	280·1	438	373·0
359	151·6	385	208·3	412	283 2	439	376·8
		386	210·8	413	286·3		
360	153·6	387	213·3	414	289·5	440	380·7
361	155·5	388	215·8	415	292·7	441	384·6
362	157·5	389	218·3	416	295·9	442	388·5
363	159·5			417	299·1	443	392·4
364	161·5	390	220·9	418	302·3	444	396·4
365	163·5	391	223·5	419	305·6	445	400·4
366	165·5	392	226·1			446	404·4
367	167·6	393	228·7	420	308·9	447	408·4
368	169 7	394	231·4	421	312·3	448	412·5
369	171·8	395	234·0	422	315·6	449	416 6
		396	236·7	423	319·0		
370	173 9	397	239·4	424	322·4	450	420·7
371	176·1	398	242·2	425	325·9		
372	178·2	399	245·0	426	329·3		

CIRCUMFERENCES AND AREAS.

Circles from $\frac{1}{16}$ in. to 100 in. diameter.

Diar.	Circumf.	Area.	Diar.	Circumf.	Area.
$\frac{1}{16}$	·1963	·00307	$2\frac{11}{16}$	8·4430	5·6723
$\frac{1}{8}$	·3927	·01227	$2\frac{3}{4}$	8·6394	5·9395
$\frac{3}{16}$	·5890	·02761	$2\frac{13}{16}$	8·8357	6·2126
$\frac{1}{4}$	·7854	·04909	$2\frac{7}{8}$	9·0321	6·4918
$\frac{5}{16}$	·9817	·07670	$2\frac{15}{16}$	9·2284	6·7772
$\frac{3}{8}$	1·1781	·1104	3	9·4248	7·0685
$\frac{7}{16}$	1·3744	·1503	$3\frac{1}{16}$	9·6211	7·3662
$\frac{1}{2}$	1·5708	·1963	$3\frac{1}{8}$	9·8175	7·6699
$\frac{9}{16}$	1·7771	·2485	$3\frac{3}{16}$	10·014	7·9798
$\frac{5}{8}$	1·9635	·3068	$3\frac{1}{4}$	10·210	8·2957
$\frac{11}{16}$	2·1598	·3712	$3\frac{5}{16}$	10·406	8·6180
$\frac{3}{4}$	2·3562	·4417	$3\frac{3}{8}$	10·602	8·9462
$\frac{13}{16}$	2·5525	·5185	$3\frac{7}{16}$	10·799	9·2807
$\frac{7}{8}$	2·7489	·6013	$3\frac{1}{2}$	10·995	9·6211
$\frac{15}{16}$	2·9452	·6903	$3\frac{9}{16}$	11·191	9·9680
1	3·1416	·7854	$3\frac{5}{8}$	11·388	10·320
$1\frac{1}{16}$	3·3379	·8866	$3\frac{11}{16}$	11·584	10·679
$1\frac{1}{8}$	3·5343	·9940	$3\frac{3}{4}$	11·781	11·044
$1\frac{3}{16}$	3·7306	1·1075	$3\frac{13}{16}$	11·977	11·416
$1\frac{1}{4}$	3·9270	1·2271	$3\frac{7}{8}$	12·173	11·793
$1\frac{5}{16}$	4·1233	1·3530	$3\frac{15}{16}$	12·369	12·177
$1\frac{3}{8}$	4·3197	1·4848	4	12·566	12·566
$1\frac{7}{16}$	4·5160	1·6229	$4\frac{1}{16}$	12·762	12·962
$1\frac{1}{2}$	4·7124	1·7671	$4\frac{1}{8}$	12·959	13·364
$1\frac{9}{16}$	4·9087	1·9175	$4\frac{3}{16}$	13·155	13·772
$1\frac{5}{8}$	5·1051	2·0739	$4\frac{1}{4}$	13·351	14·186
$1\frac{11}{16}$	5·3014	2·2365	$4\frac{5}{16}$	13·547	14·606
$1\frac{3}{4}$	5·4978	2·4052	$4\frac{3}{8}$	13·744	15·033
$1\frac{13}{16}$	5·6941	2·5800	$4\frac{7}{16}$	13·940	15·465
$1\frac{7}{8}$	5·8905	2·7611	$4\frac{1}{2}$	14·137	15·904
$1\frac{15}{16}$	6·0868	2·9483	$4\frac{9}{16}$	14·333	16·349
2	6·2832	3·1416	$4\frac{5}{8}$	14·529	16·800
$2\frac{1}{16}$	6·4795	3·3380	$4\frac{11}{16}$	14·725	17·257
$2\frac{1}{8}$	6·6759	3·5465	$4\frac{3}{4}$	14·922	17·720
$2\frac{3}{16}$	6·8722	3·7584	$4\frac{13}{16}$	15·119	18·190
$2\frac{1}{4}$	7·0686	3·9760	$4\frac{7}{8}$	15·315	18·665
$2\frac{5}{16}$	7·2649	4·2000	$4\frac{15}{16}$	15·511	19·147
$2\frac{3}{8}$	7·4613	4·4302	5	15·708	19·635
$2\frac{7}{16}$	7·6576	4·7066	$5\frac{1}{16}$	15·904	20·129
$2\frac{1}{2}$	7·8540	4·9087	$5\frac{1}{8}$	16·100	20·629
$2\frac{9}{16}$	8·0503	5·1573	$5\frac{3}{16}$	16·296	21·135
$2\frac{5}{8}$	8·2467	5·4119	$5\frac{1}{4}$	16·493	21·647

Diar.	Circumf.	Area.	Diar.	Circumf.	Area.
$5\frac{5}{16}$	16·689	22·166	10	31·416	78·540
$5\frac{3}{8}$	16·886	22·690	$10\frac{1}{8}$	31·808	80·515
$5\frac{7}{16}$	17·082	23·221	$10\frac{1}{4}$	32·201	82·516
$5\frac{1}{2}$	17·278	23·758	$10\frac{3}{8}$	32·594	84·540
$5\frac{9}{16}$	17·474	24·301	$10\frac{1}{2}$	32·986	86·590
$5\frac{5}{8}$	17·671	24·850	$10\frac{5}{8}$	33·379	88·664
$5\frac{11}{16}$	17·867	25·406	$10\frac{3}{4}$	33·772	90·762
$5\frac{3}{4}$	18·064	25·967	$10\frac{7}{8}$	34·164	92·885
$5\frac{13}{16}$	18·261	26·535	11	34·558	95·033
$5\frac{7}{8}$	18·457	27·108	$11\frac{1}{8}$	34·950	97·205
$5\frac{15}{16}$	18·653	27·688	$11\frac{1}{4}$	35·343	99·402
6	18·849	28·274	$11\frac{3}{8}$	35·735	101·623
$6\frac{1}{8}$	19·242	29·464	$11\frac{1}{2}$	36·128	103·869
$6\frac{1}{4}$	19·635	30·679	$11\frac{5}{8}$	36·521	106·139
$6\frac{3}{8}$	20·027	31·919	$11\frac{3}{4}$	36·913	108·434
$6\frac{1}{2}$	20·420	33·183	$11\frac{7}{8}$	37·306	110·753
$6\frac{5}{8}$	20·813	34·471	12	37·699	113·097
$6\frac{3}{4}$	21·205	35·784	$12\frac{1}{8}$	38·091	115·466
$6\frac{7}{8}$	21·598	37·122	$12\frac{1}{4}$	38·484	117·859
7	21·991	38·484	$12\frac{3}{8}$	38·877	120·276
$7\frac{1}{8}$	22·383	39·871	$12\frac{1}{2}$	39·270	122·718
$7\frac{1}{4}$	22·776	41·282	$12\frac{5}{8}$	39·662	125·184
$7\frac{3}{8}$	23·169	42·718	$12\frac{3}{4}$	40·055	127·676
$7\frac{1}{2}$	23·562	44·178	$12\frac{7}{8}$	40·448	130·192
$7\frac{5}{8}$	23·954	45·663	13	40·840	132·732
$7\frac{3}{4}$	24·347	47·173	$13\frac{1}{8}$	41·233	135·297
$7\frac{7}{8}$	24·740	48·707	$13\frac{1}{4}$	41·626	137·886
8	25·132	50·265	$13\frac{3}{8}$	42·018	140·500
$8\frac{1}{8}$	25·515	51·848	$13\frac{1}{2}$	42·411	143·139
$8\frac{1}{4}$	25·918	53·456	$13\frac{5}{8}$	42·804	145·802
$8\frac{3}{8}$	26·310	55·088	$13\frac{3}{4}$	43·197	148·489
$8\frac{1}{2}$	26·703	56·745	$13\frac{7}{8}$	43·589	151·201
$8\frac{5}{8}$	27·096	58·426	14	43·982	153·938
$8\frac{3}{4}$	27·489	60·132	$14\frac{1}{8}$	44·375	156·699
$8\frac{7}{8}$	27·881	61·862	$14\frac{1}{4}$	44·767	159·485
9	28·274	63·617	$14\frac{3}{8}$	45·160	162·295
$9\frac{1}{8}$	28·667	65·396	$14\frac{1}{2}$	45·553	165·130
$9\frac{1}{4}$	29·059	67·200	$14\frac{5}{8}$	45·945	167·989
$9\frac{3}{8}$	29·452	69·029	$14\frac{3}{4}$	46·338	170·873
$9\frac{1}{2}$	29·845	70·882	$14\frac{7}{8}$	46·731	173·782
$9\frac{5}{8}$	30·237	72·759	15	47·124	176·715
$9\frac{3}{4}$	30·630	74·662	$15\frac{1}{8}$	47·516	179·672
$9\frac{7}{8}$	31·023	76·588	$15\frac{1}{4}$	47·909	182·654

Diar.	Circumf.	Area.	Diar.	Circumf.	Area.
15⅜	48·302	185·661	20¾	65·188	338·163
15½	48·694	188·692	20⅞	65·580	342·250
15⅝	49·087	191·748	21	65·973	346·361
15¾	49·480	194·828	21⅛	66·366	350·497
15⅞	49·872	197·933	21¼	66·759	354·657
16	50·265	201·062	21⅜	67·151	358·841
16⅛	50·658	204·216	21½	67·544	363·051
16¼	51·051	207·394	21⅝	67·937	367·284
16⅜	51·443	210·597	21¾	68·329	371·543
16½	51·836	213·825	21⅞	68·722	375·826
16⅝	52·229	217·077	22	69·115	380·133
16¾	52·621	220·353	22⅛	69·507	384·465
16⅞	53·014	223·654	22¼	69·900	388·822
17	53·407	226·980	22⅜	70·293	393·203
17⅛	53·799	230·330	22½	70·686	397·608
17¼	54·192	233·705	22⅝	71·078	402·038
17⅜	54·585	237·104	22¾	71·471	406·493
17½	54·978	240·528	22⅞	71·864	410·972
17⅝	55·370	243·977	23	72·256	415·476
17¾	55·763	247·450	23⅛	72·649	420·004
17⅞	56·156	250·947	23¼	73·042	424·557
18	56·548	254·469	23⅜	73·434	429·135
18⅛	56·941	258·016	23½	73·827	433·731
18¼	57·334	261·587	23⅝	74·220	438·363
18⅜	57·726	265·182	23¾	74·613	443·014
18½	58·119	268·803	23⅞	75·005	447·699
18⅝	58·512	272·447	24	75·398	452·390
18¾	58·905	276·117	24⅛	75·791	457·115
18⅞	59·297	279·811	24¼	76·183	461·864
19	59·690	283·529	24⅜	76·576	466·638
19⅛	60·083	287·272	24½	76·969	471·436
19¼	60·475	291·039	24⅝	77·361	476·259
19⅜	60·868	294·831	24¾	77·754	481·106
19½	61·261	298·648	24⅞	78·147	485·978
19⅝	61·653	302·489	25	78·540	490·875
19¾	62·046	306·355	25⅛	78·932	495·796
19⅞	62·439	310·245	25¼	79·325	500·741
20	62·832	314·160	25⅜	79·718	505·711
20⅛	63·224	318·099	25½	80·110	510·706
20¼	63·617	322·063	25⅝	80·503	515·725
20⅜	64·010	326·051	25¾	80·896	520·769
20½	64·402	330·064	25⅞	81·288	525·837
20⅝	64·795	334·101	26	81·681	530·930

Diar.	Circumf.	Area.	Diar.	Circumf.	Area.
26⅛	82·074	536·047	31½	98·968	779·313
26¼	82·467	541·189	31⅝	99·353	785·510
26⅜	82·859	546·356	31¾	99·745	791·732
26½	83·252	551·547	31⅞	100·138	797·978
26⅝	83·645	556·762	32	100·531	804·249
26¾	84·037	562·002	32⅛	100·924	810·545
26⅞	84·430	567·267	32¼	101·316	816·865
27	84·823	572·556	32⅜	101·709	823·209
27⅛	85·215	577·870	32½	102·102	829·578
27¼	85·608	583·208	32⅝	102·494	835·972
27⅜	86·001	588·571	32¾	102·887	842·390
27½	86·394	593·958	32⅞	103·280	848·833
27⅝	86·786	599·370	33	103·672	855·30
27¾	87·179	604·807	33⅛	104·055	861·79
27⅞	87·572	610·268	33¼	104·458	868·30
28	87·964	615·753	33⅜	104·850	874·84
28⅛	88·357	621·263	33½	105·243	881·41
28¼	88·750	626·798	33⅝	105·636	888·00
28⅜	89·142	632·357	33¾	106·029	894·61
28½	89·535	637·941	33⅞	106·421	901·25
28⅝	89·928	643·594	34	106·814	907·92
28¾	90·321	649·182	34⅛	107·207	914·61
28⅞	90·713	654·839	34¼	107·599	921·32
29	91·106	660·521	34⅜	107·992	928·06
29⅛	91·499	666·227	34½	108·385	934·82
29¼	91·891	671·958	34⅝	108·777	941·60
29⅜	92·284	677·714	34¾	109·170	948·41
29½	92·677	683·494	34⅞	109·563	955·25
29⅝	93·069	689·298	35	109·956	962·11
29¾	93·462	695·128	35⅛	110·348	968·99
29⅞	93·855	700·981	35¼	110·741	975·90
30	94·248	706·860	35⅜	111·134	982·84
30⅛	94·640	712·762	35½	111·526	989·80
30¼	95·033	718·690	35⅝	111·919	996·78
30⅜	95·426	724·641	35¾	112·312	1003·78
30½	95·818	730·618	35⅞	112·704	1010·82
30⅝	96·211	736·619	36	113·097	1017·88
30¾	96·604	742·644	36⅛	113·490	1024·95
30⅞	96·996	748·694	36¼	113·883	1032·06
31	97·389	754·769	36⅜	114·275	1039·19
31⅛	97·782	760·868	36½	114·668	1046·35
31¼	98·175	766·992	36⅝	115·061	1053·52
31⅜	98·567	773·140	36¾	115·453	1060·73

Diar.	Circumf.	Area.	Diar.	Circumf.	Area.
36⅞	115·846	1067·95	42¼	132·732	1401·98
37	116·239	1075·21	42⅜	133·125	1410·29
37⅛	116·631	1082·48	42½	133·518	1418·62
37¼	117·024	1089·79	42⅝	133·910	1426·98
37⅜	117·417	1097·11	42¾	134·303	1435·36
37½	117·810	1104·46	42⅞	134·696	1443·77
37⅝	118·202	1111 84	43	135·088	1452·20
37¾	118·595	1119·24	43⅛	135·481	1460·65
37⅞	118·988	1126·66	43¼	135·874	1469·13
38	119·380	1134·11	43⅜	136·266	1477 63
38⅛	119·773	1141·59	43½	136·659	1486·17
38¼	120·166	1149·08	43⅝	137·052	1494·72
38⅜	120·558	1156·61	43¾	137·445	1503·30
38½	120·951	1164·15	43⅞	137·837	1511·90
38⅝	121·344	1171·73	44	138·230	1520·53
38¾	121·737	1179·32	44⅛	138·623	1529 18
38⅞	122·129	1186·94	44¼	139·015	1537·86
39	122·522	1194·59	44⅜	139·408	1546·55
39⅛	122·915	1202·26	44½	139·801	1555·28
39¼	123·307	1209·95	44⅝	140·193	1564·03
39⅜	123·700	1217·67	44¾	140·586	1572·81
39½	124·093	1225·42	44⅞	140·979	1581·61
39⅝	124·485	1233·18	45	141·372	1590·43
39¾	124·878	1240·98	45⅛	141·764	1599·28
39⅞	125·271	1248·79	45¼	142·157	1608·15
40	125·664	1256·64	45⅜	142·550	1617·04
40⅛	126·056	1264·50	45½	142·942	1625·97
40¼	126·449	1272·39	45⅝	143·335	1634·92
40⅜	126·842	1280·31	45¾	143·728	1643·89
40½	127·234	1288·25	45⅞	144·120	1652·88
40⅝	127·627	1296·21	46	144·513	1661·90
40¾	128·020	1304·20	46⅛	144·906	1670·95
40⅞	128·412	1312·21	46¼	145·299	1680·01
41	128·805	1320·25	46⅜	145·691	1689·10
41⅛	129·198	1328·32	46½	146·084	1698·23
41¼	129·591	1336·40	46⅝	146·477	1707·37
41⅜	129·983	1344·51	46¾	146·869	1716·54
41½	130·376	1352·65	46⅞	147·262	1725·73
41⅝	130·769	1360·81	47	147·655	1734·94
41¾	131·161	1369·00	47⅛	148·047	1744·18
41⅞	131·554	1377·21	47¼	148·440	1753·45
42	131·947	1385·44	47⅜	148·833	1762·73
42⅛	132·339	1393·70	47½	149·226	1772·05

Diar.	Circumf.	Area.	Diar.	Circumf.	Area.
47⅝	149·618	1781·39	56	175·929	2463·01
47¾	150·011	1790·76	56¼	176·715	2485·05
47⅞	150·404	1800·14	56½	177·500	2507·19
48	150·796	1809·56	56¾	178·285	2529·42
48⅛	151·189	1818·99	57	179·071	2551·76
48¼	151·582	1828·46	57¼	179·856	2574·19
48⅜	151·974	1837·93	57½	180·642	2596·72
48½	152·367	1847·45	57¾	181·427	2619·35
48⅝	152·760	1856·99	58	182·212	2642·08
48¾	153·153	1866·55	58¼	182·998	2664·91
48⅞	153·545	1876·13	58½	183·783	2687·83
49	153·938	1885·74	58¾	184·569	2710·85
49⅛	154·331	1895·37	59	185·354	2733·97
49¼	154·723	1905·03	59¼	186·139	2757·19
49⅜	155·116	1914·70	59½	186·925	2780·51
49½	155·509	1924·42	59¾	187·710	2803·92
49⅝	155·901	1934·15	60	188·496	2827·43
49¾	156·294	1943·91	60¼	189·281	2851·05
49⅞	156·687	1953·69	60½	190·066	2874·76
50	157·080	1963·50	60¾	190·852	2898·56
50¼	157·865	1983·18	61	191·637	2922·47
50½	158·650	2002·96	61¼	192·423	2946·47
50¾	159·436	2022·84	61½	193·208	2970·57
51	160·221	2042·82	61¾	193·993	2994·77
51¼	161·007	2062·90	62	194·779	3019·07
51½	161·792	2083·07	62¼	195·564	3043·47
51¾	162·577	2103·35	62½	196·350	3067·96
52	163·363	2123·72	62¾	197·135	3092·56
52¼	164·148	2144·19	63	197·920	3117·25
52½	164·934	2164·75	63¼	198·706	3142·04
52¾	165·719	2185·42	63½	199·491	3166·92
53	166·504	2206·18	63¾	200·277	3191·91
53¼	167·290	2227·05	64	201·062	3216·99
53½	168·075	2248·01	64¼	201·847	3242·17
53¾	168·861	2269·06	64½	202·633	3267·46
54	169·646	2290·22	64¾	203·418	3292·83
54¼	170·431	2311·48	65	204·204	3318·31
54½	171·217	2332·83	65¼	204·989	3343·88
54¾	172·002	2354·28	65½	205·774	3369·56
55	172·788	2375·83	65¾	206·560	3395·33
55¼	173·573	2397·48	66	207·345	3421·19
55½	174·358	2419·22	66¼	208·131	3447·16
55¾	175·144	2441·07	66½	208·916	3473·23

Diar.	Circumf.	Area.	Diar.	Circumf.	Area.
66¾	209·701	3499·39	77½	243·474	4717·30
67	210·487	3525·66	77¾	244·259	4747·79
67¼	211·272	3552·01	78	245·044	4778·36
67½	212·058	3578·47	78¼	245·830	4809·05
67¾	212·843	3605·03	78½	246·615	4839·83
68	213·628	3631·68	78¾	247·401	4870·70
68¼	214·414	3658·44	79	248·186	4901·68
68½	215·199	3685·29	79¼	248·971	4932·75
68¾	215·985	3712·24	79½	249·757	4963·92
69	216·770	3739·28	79¾	250·542	4995·19
69¼	217·555	3766·43	80	251·328	5026·55
69½	218·341	3793·67	80¼	252·113	5058·00
69¾	219·126	3821·02	80½	252·898	5089·58
70	219·912	3848·45	80¾	253·683	5121·22
70¼	220·697	3875·99	81	254·469	5153·00
70½	221·482	3903·63	81¼	255·254	5184·84
70¾	222·268	3931·36	81½	256·040	5216·82
71	223·053	3959·19	81¾	256·825	5248·84
71¼	223·839	3987·13	82	257·611	5281·02
71½	224·624	4015·16	82¼	258·396	5313·28
71¾	225·409	4043·28	82½	259·182	5345·62
72	226·195	4071·50	82¾	259·967	5378·04
72¼	226·980	4099·83	83	260·752	5410·61
72½	227·766	4128·25	83¼	261·537	5443·24
72¾	228·551	4156·77	83½	262·323	5476·00
73	229·336	4185·39	83¾	263·108	5508·84
73¼	230·122	4214·11	84	263·894	5541·77
73½	230·907	4242·92	84¼	264·679	5574·80
73¾	231·693	4271·83	84½	265·465	5607·95
74	232·478	4300·84	84¾	266·250	5641·16
74¼	233·263	4329·95	85	267·035	5674·51
74½	234·049	4359·16	85¼	267·821	5707·92
74¾	234·834	4388·47	85½	268·606	5741·47
75	235·620	4417·86	85¾	269·392	5775·09
75¼	236·405	4447·37	86	270·177	5808·80
75½	237·190	4476·97	86¼	270·962	5842·60
75¾	237·976	4506·67	86½	271·748	5876·55
76	238·761	4536·46	86¾	272·533	5910·52
76¼	239·547	4566·36	87	273·319	5944·68
76½	240·332	4596·35	87¼	274·104	5978·88
76¾	241·117	4626·44	87½	274·890	6013·21
77	241·903	4656·63	87¾	275·675	6047·60
77¼	242·688	4686·92	88	276·460	6082·12

Diar.	Circumf.	Area.	Diar.	Circumf.	Area.
88¼	277·245	6116·72	99¼	311·802	7736·60
88½	278·031	6151·44	99½	312·588	7775·64
88¾	278·816	6186·20	99¾	313·374	7814·76
89	279·602	6221·14	100	314·159	7853·98
89¼	280·387	6256·12	100½	315·730	7938·72
89½	281·173	6291·25	101	317·301	8011·85
89¾	281·958	6326·44	101½	318·872	8091·36
90	282·744	6361·73	102	320·442	8171·28
90¼	283·529	6399·12	102½	322·014	8251·60
90½	284·314	6432·62	103	323·584	8332·29
90¾	285·099	6468·16	103½	325·154	8413·40
91	285·885	6503·88	104	326·726	8494·37
91¼	286·670	6539·68	104½	328·296	8576·76
91½	287·456	6573·56	105	329·867	8659·01
91¾	288·242	6611·52	105½	331·438	8741·68
92	289·027	6647·61	106	333·009	8824·73
92¼	289·812	6683·80	106½	334·580	8908·20
92½	290·598	6720·07	107	336·150	8992·02
92¾	291·383	6755·40	107½	337·722	9076·24
93	292·168	6792·91	108	339·292	9160·88
93¼	292·953	6829·48	108½	340·862	9245·92
93½	293·739	6866·16	109	342·434	9331·32
93¾	294·524	6882·92	109½	344·004	9417·12
94	295·310	6939·78	110	345·575	9503·32
94¼	296·095	6976·72	110½	347·146	9589·92
94½	296·881	7013·81	111	348·717	9676·89
94¾	297·666	7050·92	111½	350·288	9764·28
95	298·452	7088·22	112	351·858	9852·03
95¼	299·237	7125·56	112½	353·430	9940·20
95½	300·022	7163·04	113	355·000	10028·75
95¾	300·807	7200·56	113½	356·570	10117·68
96	301·593	7238·23	114	358·142	10207·03
96¼	302·378	7275·96	114½	359·712	10296·76
96½	302·164	7313·84	115	361·283	10386·89
96¾	303·948	7351·72	115½	362·854	10477·40
97	304·734	7389·81	116	364·425	10568·32
97¼	305·520	7427·96	116½	365·996	10659·64
97½	306·306	7474·20	117	367·566	10751·32
97¾	307·090	7504·52	117½	369·138	10843·40
98	307·876	7542·96	118	370·708	10935·88
98¼	308·662	7581·48	118½	372·278	11028·76
98½	309·446	7620·12	119	373·849	11122·02
98¾	310·232	7658·80	119½	375·420	11215·68
99	311·018	7697·69	120	376·991	11309·73

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